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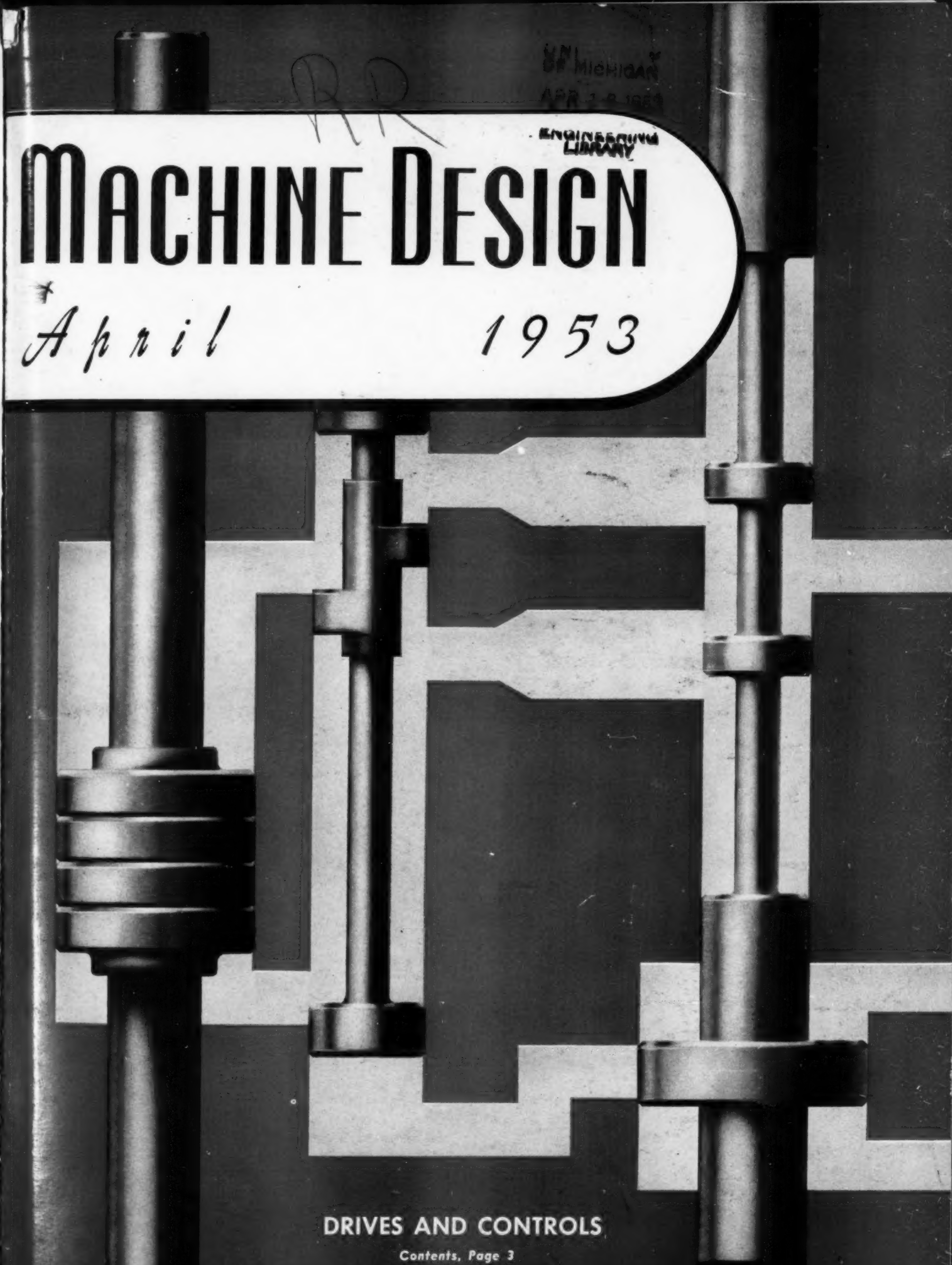
MACHINE DESIGN

April 1953



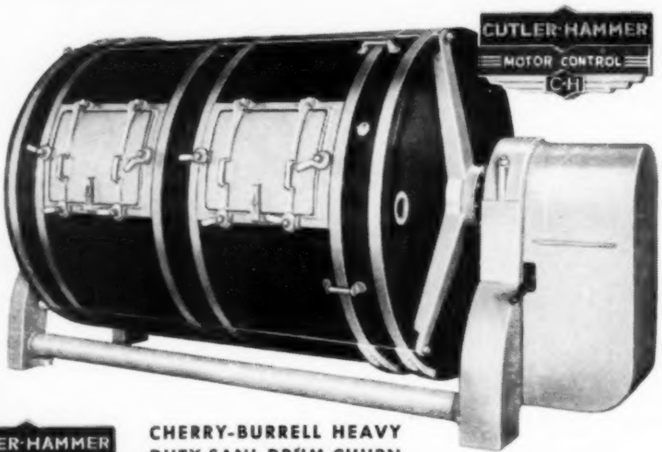
DRIVES AND CONTROLS

Contents, Page 3

CUTLER·HAMMER MOTOR CONTROL C·H

Choice of the leaders

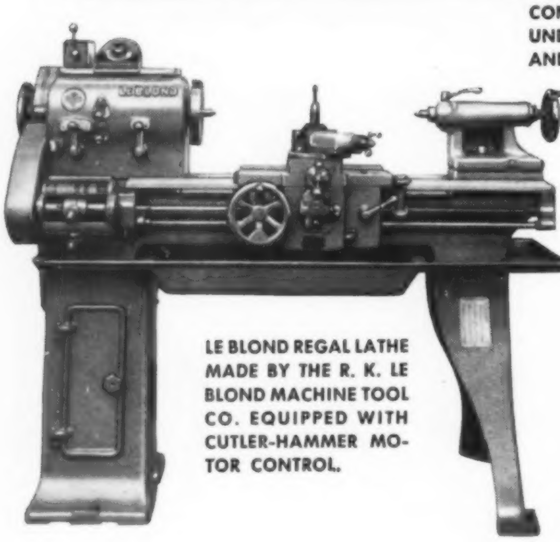
The Mark of Better Machines



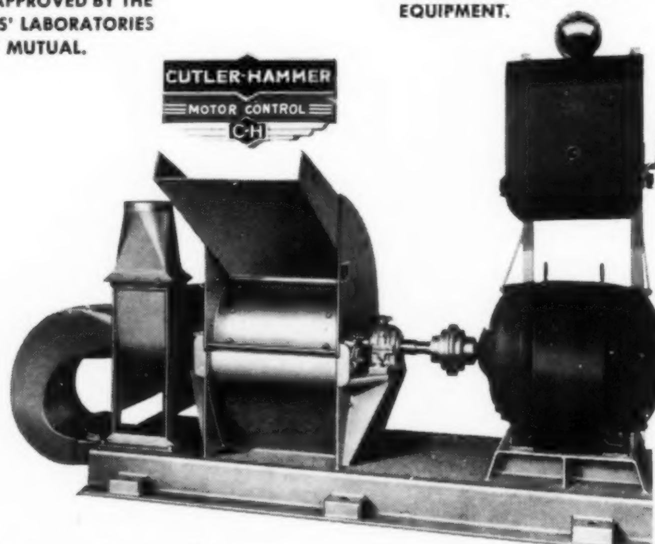
CHERRY-BURRELL HEAVY DUTY SANI-DRUM CHURN MADE BY CHERRY-BURRELL CORPORATION AND EQUIPPED WITH CUTLER-HAMMER MOTOR CONTROL.

FAIRBANKS-MORSE TURBINE FIRE PUMP EQUIPPED WITH CUTLER-HAMMER FIRE PUMP CONTROL AS APPROVED BY THE UNDERWRITERS' LABORATORIES AND FACTORY MUTUAL.

HEIL MODEL 24-24 HAMMERMILL MADE BY THE HEIL CO. CUTLER-HAMMER MOTOR CONTROL IS FURNISHED AS STANDARD EQUIPMENT.



LE BLOND REGAL LATHE MADE BY THE R. K. LE BLOND MACHINE TOOL CO. EQUIPPED WITH CUTLER-HAMMER MOTOR CONTROL.



The meaning is clear

In the never-ending struggle for business, there are a few manufacturers in every field who fare extraordinarily well. These manufacturers become acknowledged leaders. Their products are widely accepted as the best. Their names become "buy-words" for assured satisfaction. To any thinking person, the meaning must be clear. These companies gain preferred position only because their products *merit* approval. And to *hold* this advantage these companies *must* follow a policy of scrupulous attention to every detail that influences product performance in the user's hands. One more conclusion is inescapable. That machinery manufacturers who win and hold leadership in their fields, so continuously and repeatedly select Cutler-Hammer Motor Control for their machines speaks volumes for the dependability of this control. The meaning for you could not be clearer. Cutler-Hammer Motor Control is a profitable choice for you as well. CUTLER-HAMMER, Inc., 1310 St. Paul Ave., Milwaukee 1, Wis. *Associate:* Canadian Cutler-Hammer, Ltd., Toronto, Ontario.

MACHINE DESIGN

April 1953, Vol. 25—No. 4

THE PROFESSIONAL JOURNAL FOR ENGINEERS AND DESIGNERS

Over the Board

Every Man His Own Prospector

If all the machines, mechanical and electrical equipment and appliances in the Sears, Roebuck catalog were collected in one place the exhibit would represent a sizable sample of the work of our readers. In addition to the usual lines of motors, engines, tractors, lawn mowers, refrigerators, clothes washers, farm machinery, etc., the 1953 catalog offers an item with the intriguing title "portable scintillation counter for prospecting for radium from moving vehicles and the air."

The Engineering Story

As part of the effort to stimulate interest in engineering among youngsters, the Advertising Council has published an excellent booklet on "How Your Company Can Help Promote Engineering As a Career." Prepared at the request of the Engineering Manpower Commission of Engineers Joint Council, the booklet includes a chapter entitled "The Engineering Story" which highlights the advantages of being an engineer. As an established engineer, you should be interested in knowing why your choice of a career was such a happy one, so here are a few excerpts:

"Engineers do interesting and exciting work. More than any other group, engineers are men who dream (in a practical way) and actually make their dreams come true—for the benefit of all.

"Engineers do useful, necessary work. And what will be new and good and useful in the years ahead will in great part be the engineer's to discover, invent, develop and produce.

"The engineering profession offers a tremendous variety of work to choose from. There is an engineering position suited to about every temperament, every talent, every taste.

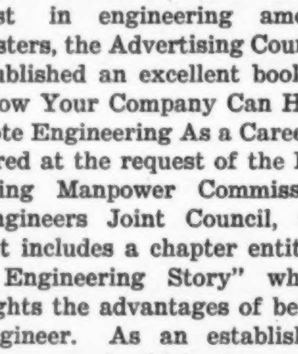
"Engineers are equipped with a marketable ability. For years to come engineers will have something industry needs, wants, and is ready to buy—specialized, technical training. A marketable ability is the only true security.

"Engineers have limitless opportunities. The man with an engineering background has a great future open to him and finds many paths leading to that future."

Some useful thoughts to leave with that promising young man in your family or neighborhood?

This Month's Cover

Many of our front covers are somewhat fanciful; others embody simplicity in the extreme. This month we are giving you a rather specific subject to look at and figure out. We wanted to symbolize the theme of this special "Drives and Controls" issue, and at the same time recognize Hadekel's fundamental article on hydraulic circuits which is its principal feature, Page 185. What you see is an adaptation of an actual circuit, which Penton artist George Farnsworth found in a schematic view of a hydraulic diesel engine governor made by Dale Hydraulic Controls Inc. Just to show that hydraulics offer only one of several ways of governing an engine or other prime mover, you will find an electronic governor announced on Page 276.



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MACHINE DESIGN
APRIL 1953

Facts or Prejudice?

IT IS IN the nature of man to be partisan. Once his allegiance has been given to a team, political party, lodge, or what have you, he is in large degree relieved of further critical thought. His emotions and the attitude of his fellow partisans create a highly subjective state of mind that is uncomplicated by mental turmoil.

Of course, engineers with their analytical minds and objective approach display none of these human frailties. Or do they? Try criticizing (constructively of course) a pet theory or design which a group of engineers has developed and see how thin is the objective veneer that covers the emotions.

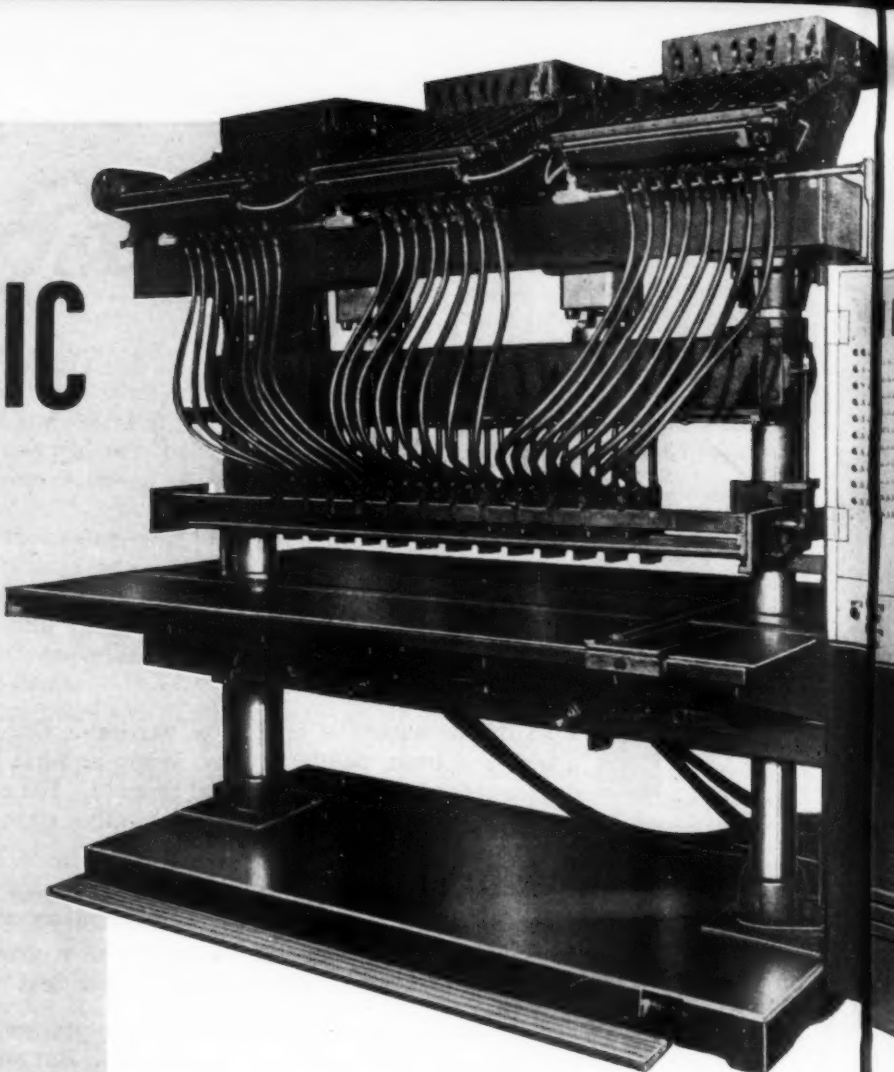
When applied to human activities like sports and politics the emotional, partisan, subjective approach is to be expected. But when scientifically trained people who should know better allow prejudice to sway their judgment in engineering matters it is a serious reflection on their good sense.

In no branch of design engineering is the temptation to let prejudice overrule judgment greater than in the selection of machine drives and controls. Partisan spirit is rife, which is only natural among those who design and manufacture for sale particular systems and components—mechanical, electrical, electronic, hydraulic, pneumatic, and so forth. But the design engineer who buys these systems and components for controlling and driving machines of his own design cannot afford the luxury of prejudice. Each new development must be reassessed and judged on its merits regardless of previous experience with similar systems or components or even with their manufacturers.

Throughout the months and years, and with extra emphasis each April issue, MACHINE DESIGN presents as well-rounded a picture of the entire machine drives and controls field as the editors and advertisers can bring together. Back of the information in these pages stand the resources of entire industries and the results of research and development costing many millions of dollars. With such powerful aids to judgment the design engineer has at hand the means for arriving at sensible, objective decisions—decisions made on the basis of facts rather than prejudice. The competent designer knows his facts and how to use them. There is no place for prejudice.

Colin Carmichael

EDITOR



AUTOMATIC RELAY CONTROL

Magnetic relays, often used only as auxiliary equipment, also possess definite advantages as major units in sequencing and program controls. Possible areas of usage are broad, as shown by the versatile telephone-type relay control discussed

By Eric Brooke
Planning Engineer
The North Electric Mfg. Co.
Galion, Ohio

IN THE past few years, much has been said about machines that are supposed to think, or have a memory. Actually, at the present time, this stage of development has not been reached, and machines are only capable of working from material supplied from a human brain doing the thinking. The so-called "memory" consists of ability to record and store information in a varying amount, according to the size of the storage mechanism provided, and then utilizing this stored information for future operations. These operations can be incorporated in machines by mechanical or electrical methods, or a combination of both.

The trend in manufacturing design has been to make machines more and more automatic. Large production machines designed for a single item have mechanical controls built in for each step in the production procedure. However, not all production procedures can utilize a single-application machine efficiently, and there is a demand for machines designed so that the program can be changed to permit a variety of production setups, and so utilize the machine time efficiently.

In these cases, electrical controls, *Fig.* 1, have some definite advantages. Mechanical connections, difficult to line up and maintain, can be eliminated. Control-

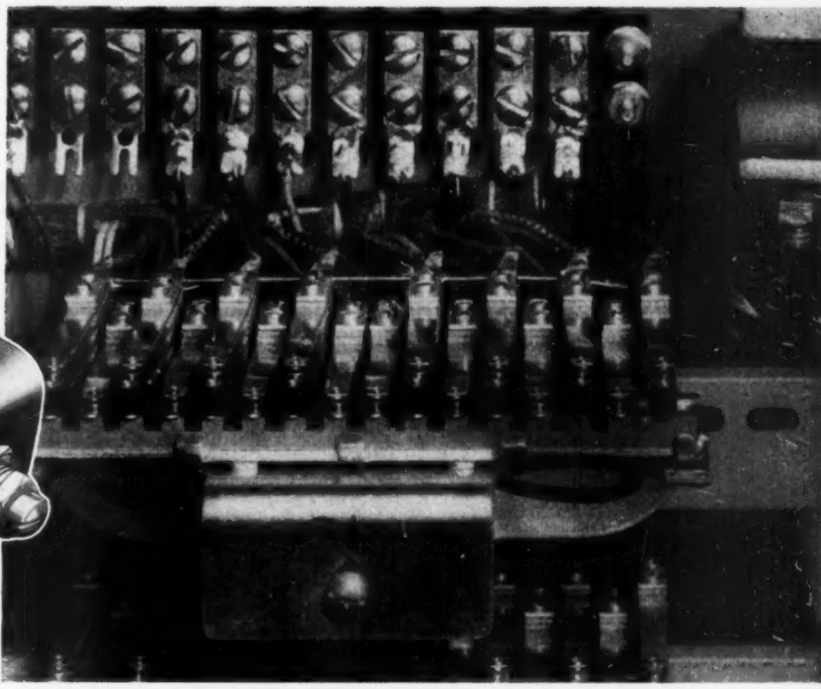


Fig. 1 — Left — Nailing machine utilizes versatile relay controls for setting up nail patterns and sequence of operations

Fig. 2—Center—Telephone type relay with three Form C contacts, having a short operating time, is well-suited for machine control circuits

Fig. 3—Above—Multicontact relay with 24 Form A contacts is used as a solenoid relay in the control

ling circuits can be switched rapidly to other sequences or disconnected, permitting quick setups for different operations. While such changes can be made with mechanical controls, they usually involve the physical changing of gears, cams, indicating wheels, etc., and require the services of skilled personnel. Electrically controlled units, however, may be arranged with the circuits operating through switches on a control panel, so that semiskilled or nonskilled help can easily change to any program available in the machine. In certain simple operations, direct electrical control can be provided with limit switches, which operate or reverse electric motors that perform the necessary control or operating functions. This method, however, is restricted to a small number of predetermined functions. Where a machine must perform a number of different functions in which the various sequences can be readily changed, and a flexible arrangement of recording and advancing sequences of operation is desired, relay control presents interesting possibilities.

Using Telephone Type Relays: A relay is a magnetically operated switch for controlling one or more circuits, either locally or between remote units. Controlled circuits can involve other relays, signals, or mechanical devices. Readily adapted for machine control functions, the telephone type relay, ***Fig.*** **2**, has a long history of reliable, trouble-free operation. Such relays have an operating time of between 10 and 15 milliseconds.

When this type of relay is used in automatic telephony, instructional information is delivered by means of electrical pulses from the telephone subscriber's dial. The relays themselves establish connections in "all-relay" systems, or control the operation of the mechanical switches in other systems. Adapting this type of relay for machine control, therefore, involves only another application of its original function. In fact, a simpler arrangement is possible, because the information can be established by means of pushbuttons or switches rather than dial pulses. Only a relatively small number of sequences are required on a repetitive basis as compared to the flexible arrangement provided in a telephone exchange, where any line or trunk can connect to any other line or trunk in the office.

Relay circuits are susceptible to very little operating trouble, which can usually be classed as: (1) opens in the operating circuit, (2) contact failures, or (3) shorts in the insulation. All of these can practically be eliminated by correct design and manufacturing

techniques. Operating circuits should be carefully designed so that conductors are of adequate size and will not be damaged during machine manufacture, thus causing trouble or damage in operation. All connections should be soldered, and wire should be of a type that is not likely to be nicked during skinning operations. Coils should be of adequate size, so that unduly fine wire is not required, and windings should be properly terminated, so that strain does not occur on the fine wire ends. Flexibility should be provided where movement is likely to be encountered.

Contact trouble can be reduced to a negligible factor by providing contacts with good follow and pressure, correct material, and proper protection against spark erosion. Relatively simple dust covers protect the relay from failure due to foreign material or films collecting on the contacts, since contacts are to a large extent self-cleaning.

Preventing insulation failure is merely a matter of selecting insulating material of suitable dimensions and dielectric strength, and being sure that manufacturing processes do not damage the insulation abilities of this material. Dust covers provided for contact protection should also provide protection against shorts caused by foreign metallic particles.

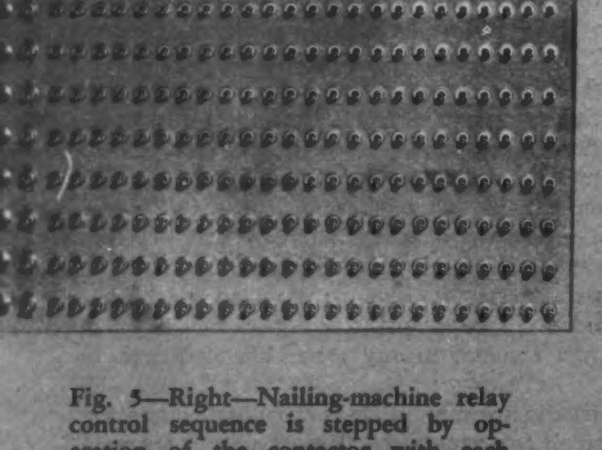
Being of sturdy design, most telephone type relays are not particularly susceptible to vibration or shock damage. If these factors are extreme, then a shock mounting for the relay control units may be necessary. Slight vibration is more of an advantage than a problem.

Current-Carrying Capacity: In designing machine control circuits using telephone type relays, consideration must be given to the current-carrying ability of the contacts, as it is somewhat limited. While AIEE specifications classify the difference between a contactor and a relay as the ability to carry more or less than 25 amperes, most telephone type relay contacts are designed for a rating of 1 ampere or less. Therefore, where heavier currents are required for actual machine control, larger power relays or contactors should be utilized. However, the number of larger units can often be reduced by careful circuit design.

Dc operation is preferable for these relay control units. While power demand is very low, use of dc current eliminates core losses present in ac-produced fields, so overall current requirements can be reduced. A dc relay can pull and hold, without chatter, a larger spring load than an equivalent-size ac relay, permitting the use of fewer relays and an easier and more straightforward circuit design. Positive control of relay operation and release speeds by means of specified adjustments are also provided, not dependent upon the frequency of the power supply. Although the provision of dc current normally necessitates a separate dc source, current demand is so low that a small, inexpensive rectifier is usually satisfactory.

While telephone type relays may be considered as somewhat "delicate" by electricians accustomed to working with larger power relays and contactors, experience has shown that they perform with remarkable reliability. The possibility of a failure due to loss of adjustment is very remote; maintenance adjustment should not be necessary, and the only maintenance

Fig. 4—Below—Control panel for the nailing machine has 10 sequences with up to 24 nails per sequence



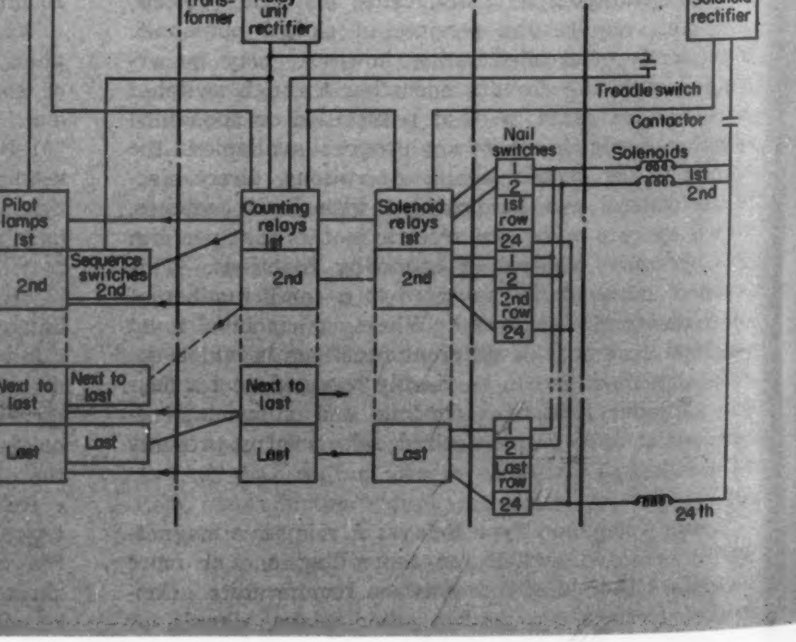


Fig. 5—Right—Nailing-machine relay control sequence is stepped by operation of the contactor with each blow of the driving head

work which should be required is the burnishing of contacts if they become dirty after a long period of use in contaminating environments.

When speed of operation does not demand electronic circuits, relay circuits have the advantage of simplicity, a simple, single low-voltage dc supply being all that is required. Electron tubes are generally limited to one control circuit per tube with one side of the control and controlled circuit common, whereas relays are available which can control a large number of independent circuits. *Fig*. 3 illustrates such a relay controlling 24 separate circuits. A relay is often necessary in the final output of electronic circuits, especially if an ac control circuit is used. With relay control, this final relay can be included in the actual controls as part of the circuit, not just an auxiliary item.

Relay Control of a Machine: An excellent example of relay control is the nailing machine manufactured by G. M. Diehl Machine Works Inc., *Fig*. 1, with a program control designed and manufactured by The North Electric Mfg. Co. This machine is designed for nailing boxes, packing cases, skids, pallets, etc., and provision has been made for a number of different nailing patterns for use in each sequence.

Relays are mounted in the rear of the control panel on the right-hand side of the machine. Controls are provided for ten different operations of up to 24 nails each. The control panel, *Fig*. 4, consists of ten pilot lamps, nine sequence switches, and ten rows of 24 nail switches each. To set up the machine for any particular program of nailing sequences, the sequence switches in consecutive rows are operated to indicate the number of sequences actually desired, and then the nail switches in each one of the rows are closed for the particular nails desired in that sequence. The machine has 24 nailing heads, each with a solenoid-controlled chute to the nail chuck. During each operation of the driving head, the solenoids release nails into the chutes for the succeeding operation, and at the end of the sequence, the controls reset and repeat the operation indefinitely.

Original design of this machine provided for a mechanical linkage and ratchet arrangement to step a rotary switch to operate the nail solenoids. However, difficulties were experienced in adjusting, maintaining and lubricating the linkage and ratchet arrangement in operation. Also, the rotary switch was not too satisfactory because of mechanical wear and erosion of the contacts. An electrically controlled magnetic stepping switch was subsequently tried but was not satisfactory for the same reasons. Application of the relay controls for these functions eliminated these difficulties.

The relay control, *Fig*. 5, consists essentially of a chain of relays stepped by each operation of the driving head. These counting relays each close the circuit to a solenoid control relay. The 24 conductors from each of these relay contacts connect through the nail switches to the 24 solenoids. During the stroke of the driving head, a contactor is operated which energizes the solenoids whose circuits have previously

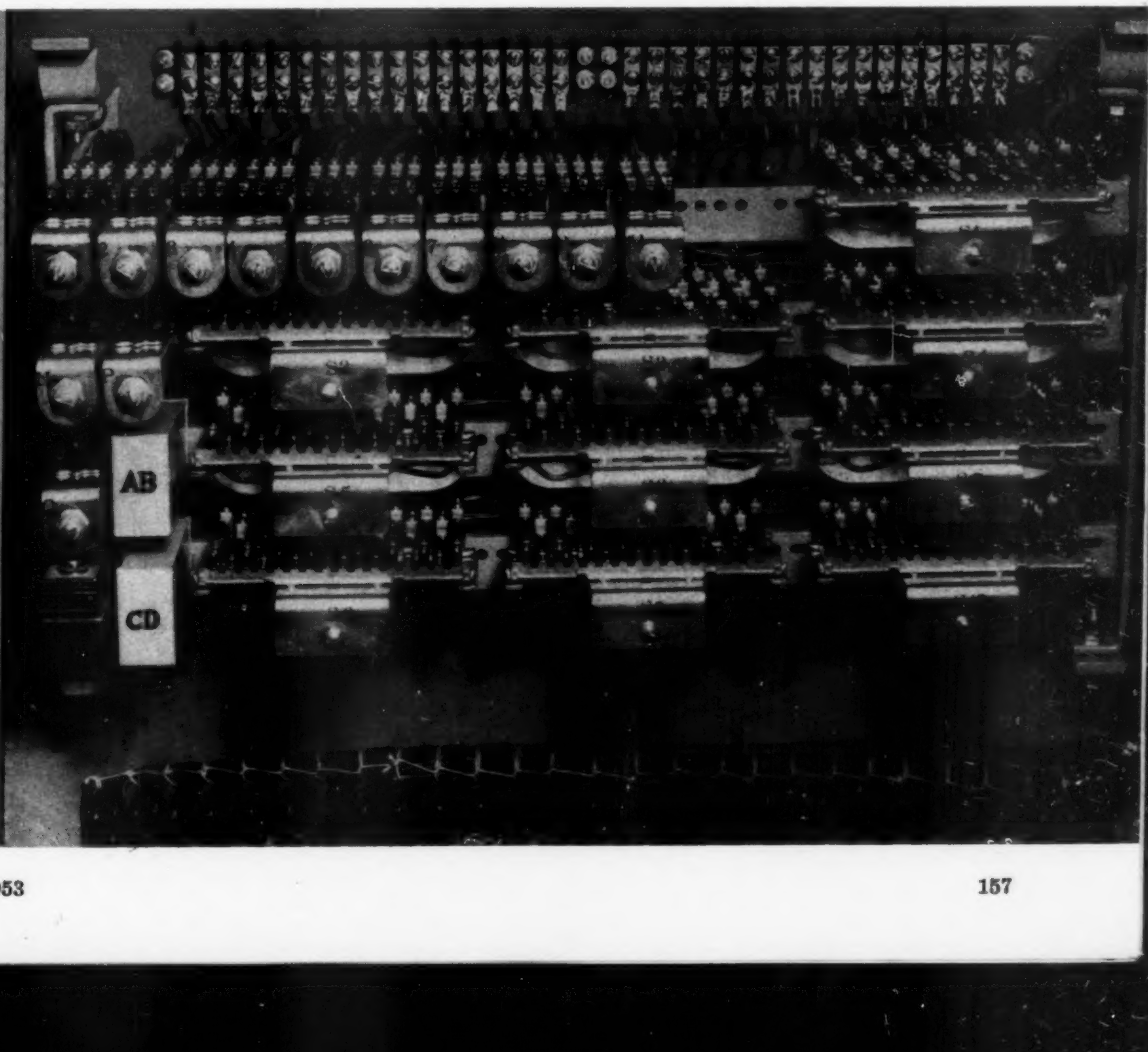


Fig. 6 — Complete relay-control unit mounted on the rear of the control panel

been prepared, by closing the appropriate nail switches, and the desired nails are delivered to the chutes. The counting relays also switch the leads to the pilot lamps to advance the light so that the number of the nail sequence which will be driven on the next operation of the driving head is indicated. The treadle-switch circuit can be arranged for individual operation, with the treadle operated for each sequence, or it can be arranged so that the machine operates continually when the treadle is depressed, each stroke of the driving head stepping the counting relays and preparing the solenoid circuits for the next sequence of operations.

The relay control unit is shown in *Fig.* 6. The control relays illustrated in *Fig.* 2 are in the top row at the left and on the left-hand side. Contact-protection networks are in the metal cans marked AB and CD beside the relays, and the small rectifier for the relay unit is in the extreme lower left. The ten solenoid relays illustrated in *Fig.* 3 occupy the balance of the mounting space. At the top are the connecting terminals to the machine. The first 24 connect direct to the solenoids; two are vacant; two are connected to the treadle switch; two are for the 6-volt ac lamp leads; two are for the 115-volt ac supply; and one is connected to the contactor. The negative side of the rectified current is also terminated on one of these binding posts so that an outside dc supply can be used if desired.

The relay unit takes little current, and a small rectifier would easily handle this load except during the operation of dc solenoids, when momentary heavy current requirements would drop the voltage to a point that could cause failure of these relays to operate or hold. By providing separate small rectifiers for the relays and the solenoids, or by operating the solenoids direct from ac current, the expense of a rectifier large enough to maintain voltage during solenoid operation is avoided in this design.

Contacts on the solenoid-control relays are adequate to carry heavy current demand, but would not last very long in service if they were required to make and break this load. By using a single large contactor for this operation, the cost of 240 heavy contacts on the solenoid relays is saved. It should be noted that the nail selection for a particular sequence is made during the preceding driving operation. This arrangement was necessary to provide sufficient time for the nails to fall from storage bins to the nail chucks. Actual relay and solenoid operation for selecting the nails was fast enough to make the selection before each stroke, but the mechanical lag in handling the nail to the chuck made this preselection feature necessary.

Summary: The application of relay control to this machine demonstrates advantages to be gained by this type of control which, in general, are: (1) ease with which control circuits can be connected to the machine with special emphasis on the possibility of remote location of the control panel; (2) flexibility possible by providing switches on the control panel for changing programs; (3) speed of operation of the relay control, which is more than adequate for most types of machine operation; and (4) simplicity and reliability of this type of control.

Points which have to be watched in designing these controls are: (1) limitations on the contact current-carrying abilities of these relays and possible use of a single large contactor to handle heavy loads; and (2) provision of an adequate power supply which might be a separate supply to prevent irregular operation caused by varying voltages.

ACKNOWLEDGEMENT

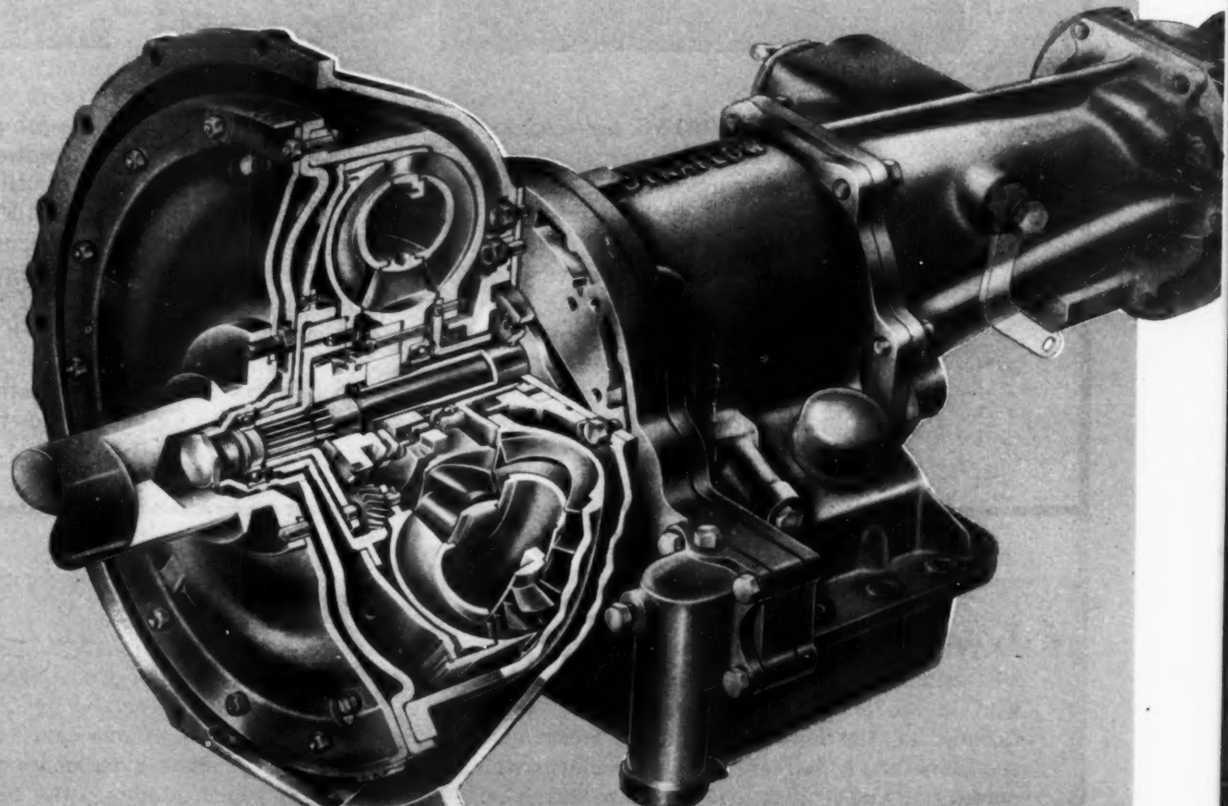
The assistance of Mr. John A. Collinge, president, G. M. Diehl Machine Works Inc. is acknowledged with appreciation.

Large Hammer Has Unusual Lubrication System

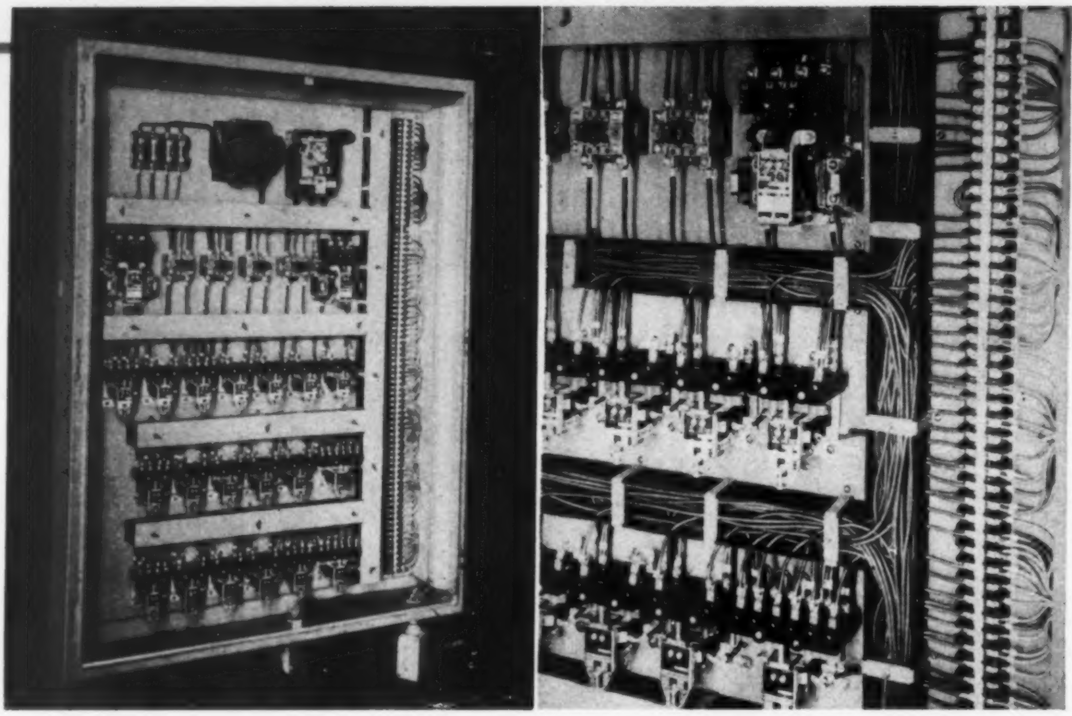
AN UNUSUAL system for lubricating the guide V's is employed in the largest steam drop hammer in this country, involving a counterbalance type of distribution valve which permits extremely sensitive control for a machine of this size. Among other features of the new design is use of a key between each of the frames and the sow block which keeps frames firm and prevents scale from working under the frame seats. Housing bolts are vertical. The bottom cylinder head, which is integral with the cylinder, is turned outside to serve as a huge dowel between the cylinder and the tie-plate. Pockets for the gland bolts are machined from the solid in the cylinder head.

Rated at 50,000 pounds, the hammer was built by Erie Foundry Co. and will be used to produce equipment for the Air Force. The completed hammer weighs over 800 tons and, when erected on its foundation, extends 16 feet below floor level and approximately 30 feet above the floor. It can be controlled by one operator using a foot treadle or hand levers.

SCANNING the field for IDEAS



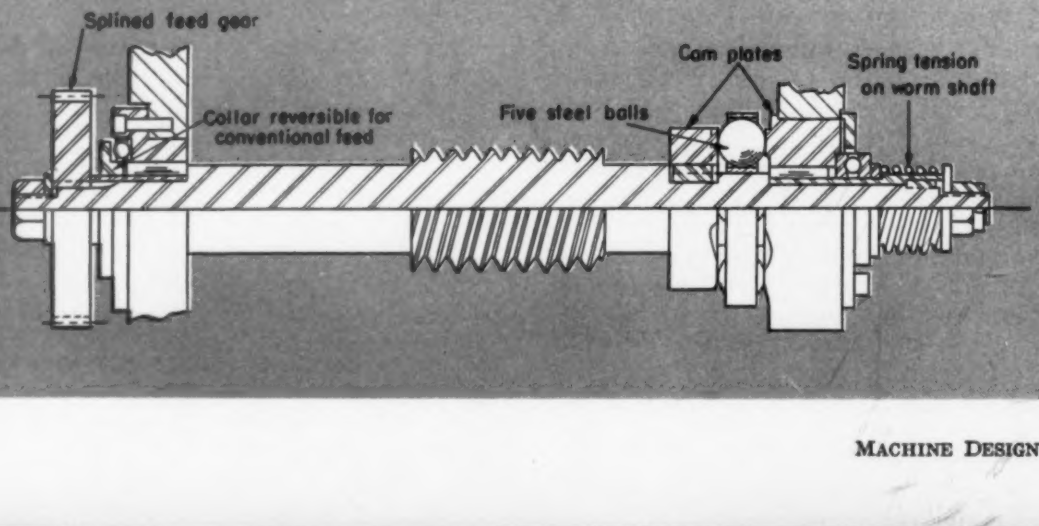
TWIN-TURBINE design of the new Buick Dynaflow torque-converter transmission increases torque output nearly 10 per cent. Addition of the secondary turbine and a planetary gearset increases torque multiplication from the 2.25-to-1 ratio of earlier units to 2.45-to-1. In operation, oil flows from the pump through the first turbine, then through the second turbine before being directed through the stator back to the pump. The first turbine drives the ring gear of the converter gearset which is connected to the output shaft through planetary gearing. The carrier and second turbine assembly is splined to the output shaft. Driving torque delivered by the first turbine is multiplied by the planetary gearing during starting but, as speed increases, this torque gradually diminishes while that of the secondary turbine increases. At cruising speed the first turbine free-wheels.



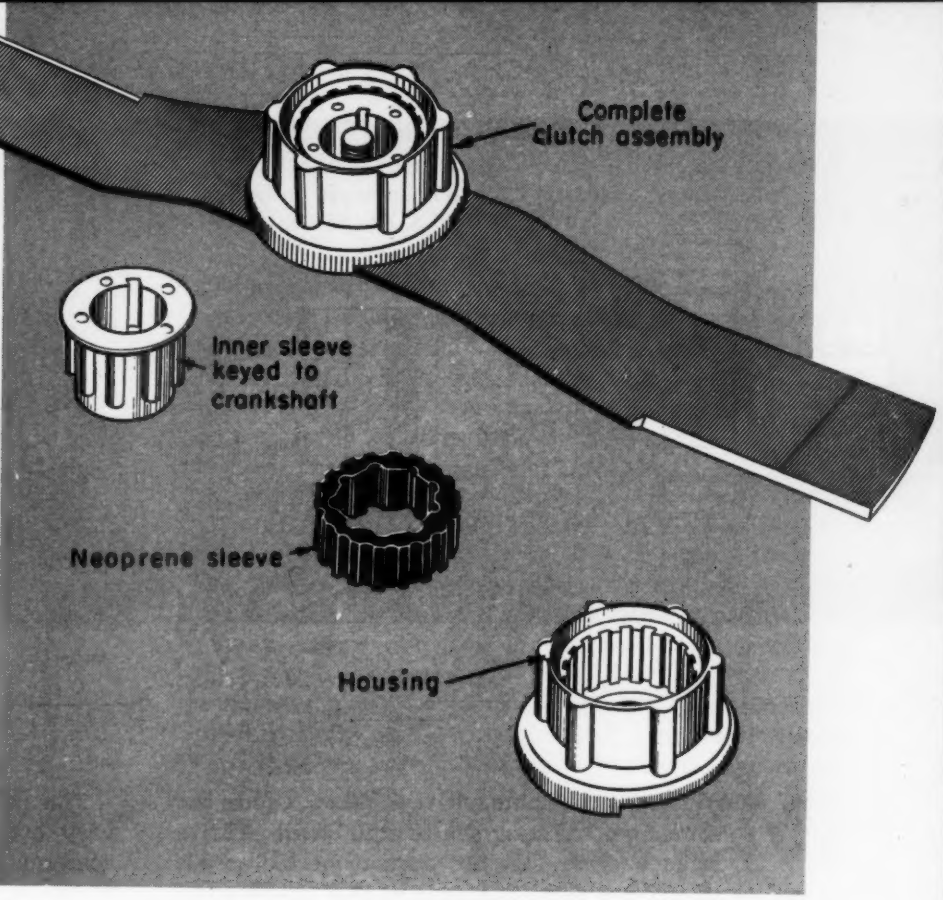
CHANNEL WIRING for machines provides a neat and workmanlike job and eliminates the need for detailed planning usually associated with conventional methods. Developed by the Snyder Tool & Engineering Co., this system utilizes special U-shaped fiber channels or ducts as main arteries for the wires. Fastened to the control panel between rows of electrical components, the channels permit direct installation of wires without special forming or tying and a regular pattern of perforations in the sides allows wires to be pulled through for connections as needed. When wiring is complete, covers are fastened over the open duct faces with wing nuts.

Besides simplifying installation, the system requires less wire and installation man-hours are reduced. Alterations and servicing can be performed without difficulty.

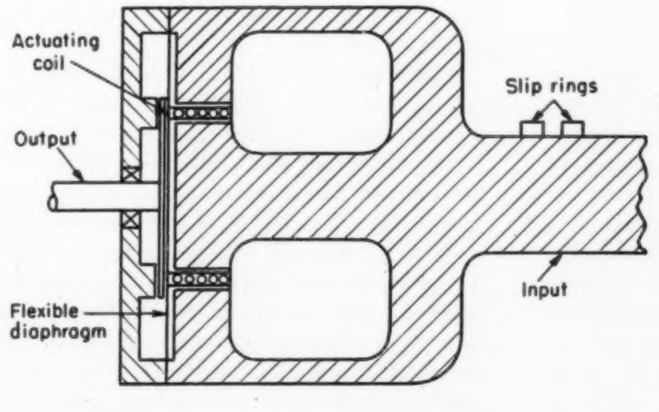
INTERMITTENT FEED of the tools on Greenlee automatics to break up chips is achieved with a novel worm drive hesitation device. Located on the lower feed-change worm shaft, the drive has a pair of keyed cam plates which operate against five steel balls to produce a lateral movement of the shaft. When riding on the inclined portion of the cams, the balls impart a lateral movement to the worm shaft which accelerates the worm wheel to produce a proportionately increased feed. As they move away from the rise of the cams, the worm shaft retracts to produce a momentary hesitation of the worm wheel, stopping the feed on all tools and breaking the chip formation. Cam motion stops but does not reverse the worm wheel.

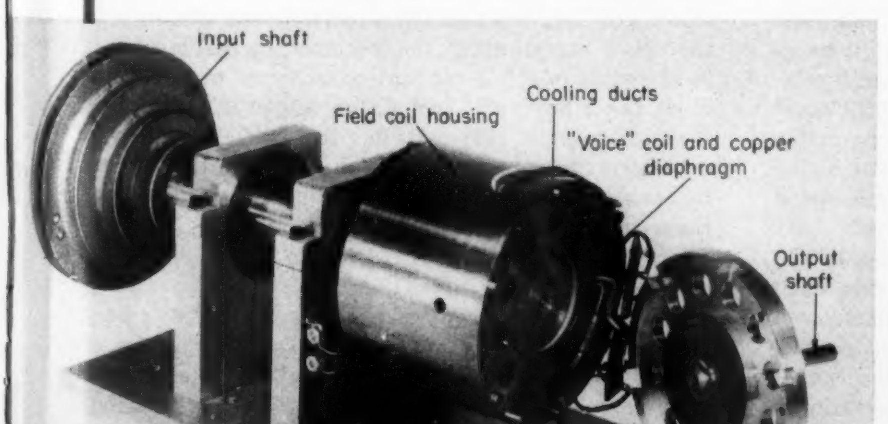


IDEAS



SLIP CLUTCH devised to utilize a notched neoprene sleeve provides the safety of shear pins along with automatic resetting. Developed by the Western Tool & Stamping Co. for a rotary mower blade drive, this clutch design consists of three concentric sleeves. Outer surface of the inner sleeve is fluted with rounded, shouldered splines and under shock or stalling loads the metal and neoprene flutings squeeze past each other to permit slip. Deformation of the synthetic sleeve absorbs the energy of impact and, when engine inertia is absorbed, the flutings automatically re-engage the drive. No adjustments are necessary and test and service runs indicate extremely long life.





LOUDSPEAKER CLUTCH design developed by the National Bureau of Standards employs the same moving-coil principle as the electrodynamic radio speaker. Fast acting, the clutch is activated by applying direct current to a coil located in a constant magnetic field. Force resulting from interaction of the coil current and the magnetic field moves the coil to press the clutch output disk against the rotating input member. In an experimental model, full output shaft torque of 10 ounce-inches, maximum, was attained in less than one-third millisecond after application of actuating voltage. Developed by Jacob Rabinow, head of the NBS Electromechanical Ordnance Division, the clutch offers advantages in instrumentation, computer, recording, photographic, and switching applications where light loads must be accelerated at extremely rapid rates.

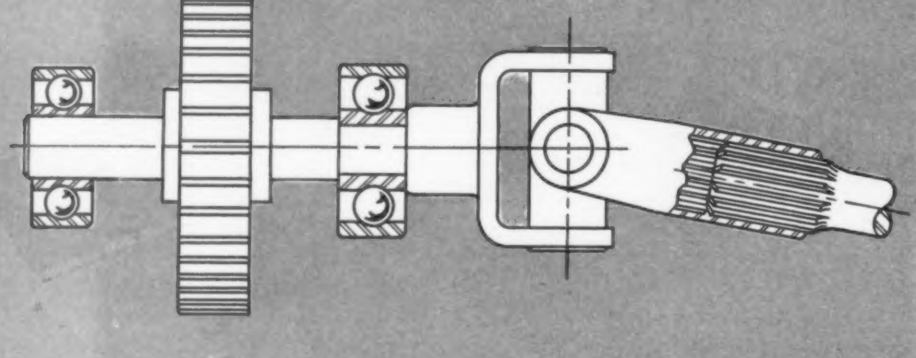


Fig. 1—Common cross-pin universal joint drive with shafts at a moderate angle

UNIVERSAL

By A. H. Rzeppa
Consultant
Gear Grinding Machine Co.
Detroit, Mich.

MANY modern machine drive systems require the coupling of rotating shafts at an angle relative to each other. In the past, moderate speeds and relatively low shaft angles gave rise to few problems. In recent years, however, speeds and working angles for such drives have steadily increased and demands for less torsional or lateral vibrations have become more urgent. As a result, a general appraisal of universal joints and their practical application in angular drive systems is highly desirable.

Plain Universal Joints: Common cross-pin type joints, originated centuries ago by Cardan and Hooke, have been developed into many forms of high mechanical perfection and have been successfully applied wherever speed and velocity were not critical, ***Fig.* 1**. Inherent in this design, however, is the undesirable feature of irregular action which cannot be eliminated without abandoning the basic design. Dynamic limitations of these simple joints are best illustrated by analysis of their mechanical performance characteristics, which are substantially alike for all such joints.

For each complete revolution of a Cardan joint operating at a specified angle, there are two positions in which the driven shaft has advanced in rotation relative to the driving shaft and two intermediary positions in which the driven shaft has lagged a similar amount. These advances and lags, alternating twice for each revolution, result in pulsating, variable speed of the driven shaft.

As the joint angle increases, the amplitudes of the pulsations increase at an even more rapid rate until they have a destructive effect upon the joint as well as the parts connected with it. At such angles this type of joint is obviously impractical. Even at small angles, slight torsional pulsations while seemingly negligible may be destructive at the higher shaft speeds common in modern machines. In any event they will be the source of objectionable vibration, noise and rapid wear, especially when the inertia of the connected rotating masses is considerable.

The performance characteristics of Cardan joints vary only slightly depending upon their particular construction, but may be substantially summarized as in the diagram of *Fig.* 2. Constant-speed rotation of the driving shaft through 360 degrees is represented by a circle with a constant vector, c, for the radius. Driven shaft variable-speed rotation, on the other hand, is represented by the superimposed concentric ellipse in which the instantaneous speeds at any given angle of rotation are indicated by the variable length of vector, v.

There are four points U at the intersection of the ellipse and circle at which the speeds of both shafts are matched. The included areas between ellipse and circle, comprise the total gain or loss of speed of the driven over the driving shaft in a typical Cardan joint, and being alike, but opposed, cancel out.

Values for speed v, acceleration α, and displacement Δ can be determined easily by the following graphical method. As mentioned constant speed is represented in a polar vector diagram by the circle described with the unit of constant velocity c as radius. Applying joint angle A, the cosine OD and secant OE will be the short and long axes of the superimposed ellipse with vectors v representing the variable speeds measured in the scale of c. A tangent constructed at any point X of the ellipse will include an angle β with a line through point X perpendicular to ray OX. The trigonometric tangent of β is a measure of the instantaneous acceleration or deceleration at X. The values range between zero at the four terminals of the major and minor axis of the ellipse to a maximum at points U.

Angular displacement of the driven relative to the driving shaft is the difference between the angles of rotation for both shafts at the same instant expressed as $\Delta = \phi - \theta$ where ϕ and θ designate the positions of the driven and driving shaft, respectively. For any joint and driving shaft angles, ϕ is found from the equation: $\tan \phi = \tan \theta / \cos A$.

In *Fig.* 2 $\tan \theta$ is shown as length FG and the $\cos A$ as OD. Division of FG by OD gives the new length

L JOINT DRIVES

Design of modern constant-velocity joints, their characteristics and application in angular drives

FH equal to tan θ. At this particular point of the ellipse the displacement angle Δ is included between rays *OG* and *OH*.

In Cartesian co-ordinates the above values of variable velocity v, acceleration α and displacement Δ are shown graphically at *Fig. 2a, b,* and *c*. Angular shaft displacements grow rapidly with increased joint angles, as may be seen in the graph of *Fig.* 3. Values at various joint angles, A, may be roughly calculated as follows: Total speed variation $= 0.033\ A^2$ per cent of constant speed; and total angular displacement $= 0.0045\ A^2$ degrees.

Constant-Velocity Joints: Primarily to overcome these deficiencies in performance, a great variety of constant velocity joints has been developed, *Fig.* 4. Such mechanisms produce true constant velocity much like a set of bevel gears in which the teeth are replaced by an intermediary member adjustable to a change of bevel angle. The characteristic feature of these joints is their symmetrical layout with respect to a third member, such as the balls and ball cage in *Fig.* 5. In most cases the value of a novel design can be immediately judged by the presence or lack of such a third member and/or the symmetry of the adjoining driving and driven members with respect to the intermediary member.

Another group of joint designs can be classed as fundamentally of the variable-speed type but include corrective or compensating parts to convert the variable into constant velocity. However, joints of this second group, due to their complicated and often sensitive structure, have had only moderate success, and are not included in this discussion.

Of the first group, the Tracta, Bendix-Weiss and Rzeppa universal joints conform to the mentioned features, namely, a driving and a driven member symmetrically arranged about a third transmitting mem-

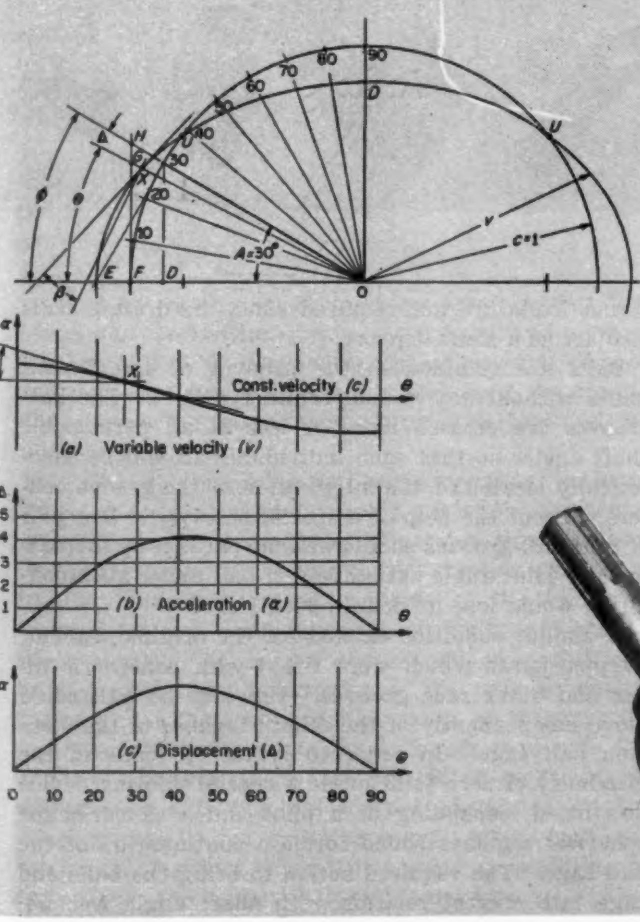


Fig. 2—Left—Polar diagram showing characteristics of constant versus cross-pin type joint rotation

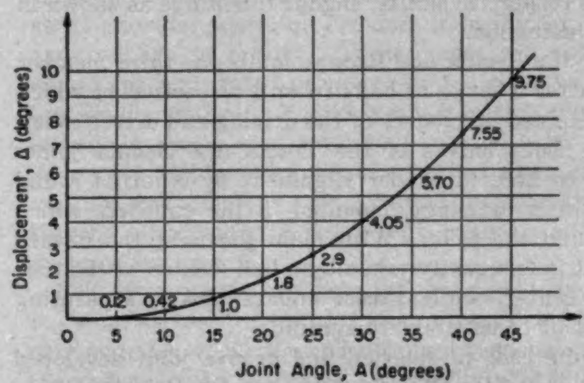


Fig. 3—Above—Chart showing degrees of advance or lag of common universal joint driven shaft at joint angles up to 45 degrees

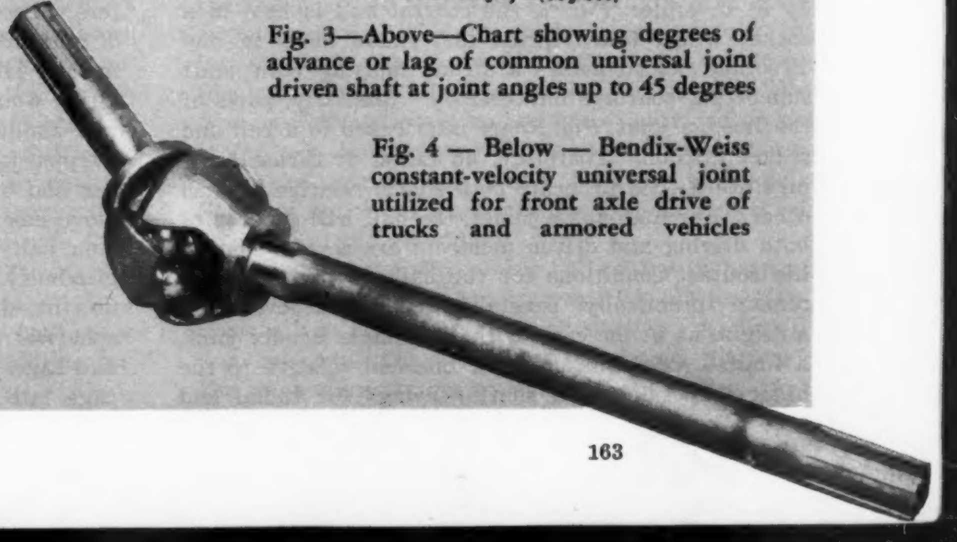


Fig. 4 — Below — Bendix-Weiss constant-velocity universal joint utilized for front axle drive of trucks and armored vehicles

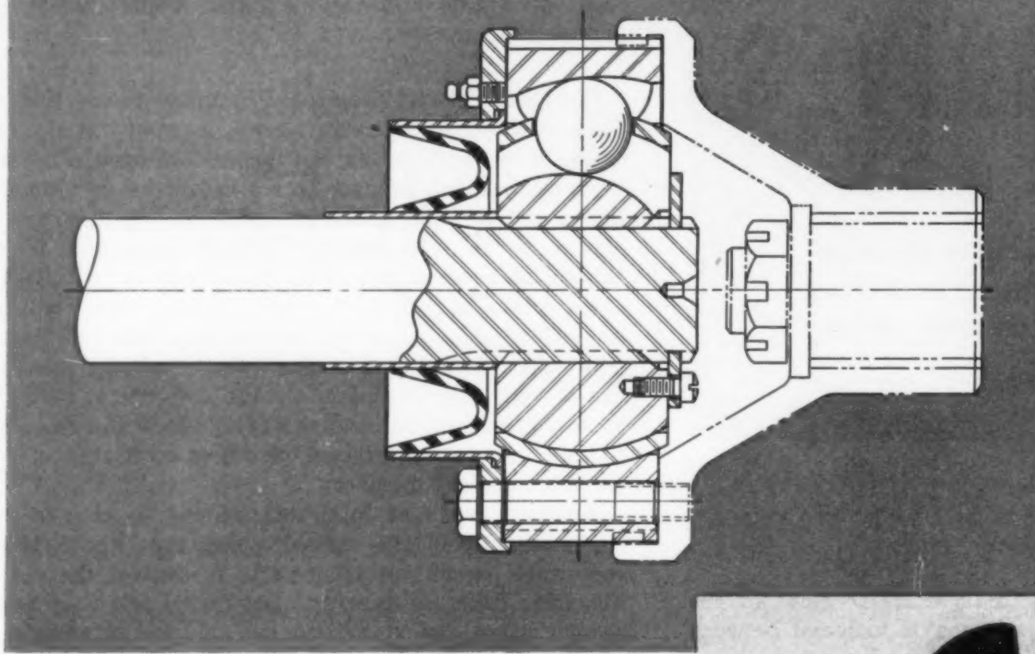
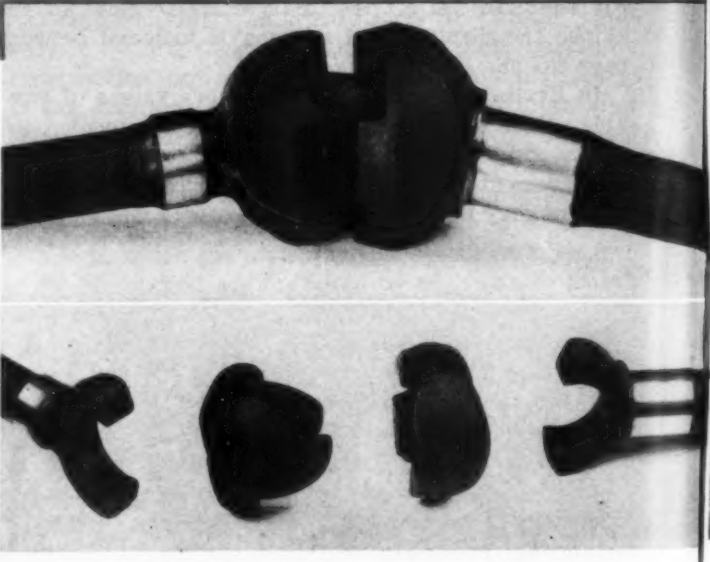


Fig. 5—Left—Propeller driveshaft with a standard constant-velocity disk joint of latest design for severe service

Fig. 6—Below—Tracta constant-velocity joint showing slotted and spigotted design somewhat akin to the Oldham coupling



ber operating in a plane which must bisect the joint angle, be perpendicular to the plane common to both shafts and located by the true joint center. Today, these requirements are too well established and known to require further analysis.

For the Tracta joint, *Fig.* 6, the third member is a hinge consisting of a slotted and a spigotted part, joined in the manner of an Oldham coupling, which can be deflected to the required joint angle and will also permit a limited amount of telescoping required for proper functioning. Two identical forks, attached to the respective shafts, engage this hinge as shown in the illustration.

For the Bendix and Rzeppa joints the third member comprises a series of balls (4 or 6 respectively) which are retained in grooves of the driving and driven members. Both halves of the Tracta and Bendix joints must be held in proper alignment by external radial and thrust bearings, mounted in the supports which surround the joint. With plain journals, the inward thrust is frequently taken by a ball located at the true joint center, while thrust collars absorb separating thrust of considerable magnitude.

In the case of the ball and groove joint developed by F. F. Miller, *Fig.* 7, the central ball is held in a socket which takes thrust and radial loads in any direction and so makes a strong housing with shaft supporting bearings unnecessary. Similarly, parts of the Rzeppa joint, *Fig.* 8, are interlocked in a ball and socket fashion. Therefore, no radial or thrust bearings are needed to locate both shafts relative to each other. For high-angle joints, the half ball-grooves of both driving and driven members are so curved that the contact conditions for the balls in their grooves remain practically constant. When grooves are straight, as in the case of the low-angle Bendix joint, a limited axial movement of one end relative to the other is possible and shaft bearings for radial and thrust loads are not required since the driving balls also act as a shaft support.

Balls are retained in the grooves of the Bendix joints without any special retainer, *Fig.* 4. The half grooves are crossed more or less at all permissible shaft angles so that each individual ball will be theoretically located at the intersection of the groove centers without the help of a special ball cage. If a pair of half ball-grooves should become parallel by increasing the joint angle excessively, a ball under such condition would lose its definite location.

A similar condition existed for the original, earlier, Rzeppa joints which were fitted with concentric inner and outer race grooves. However no ball could move independently of the others because of the common ball cage. In order to locate the balls in the proximity of zero joint angle a special piloting device was fitted, consisting of a pilot and a saucer-shape spherical segment which forms a continuation of the ball cage. The required action to bring the balls and cage into correct relation with shaft angle was ob-

Fig. 7—Right—Ball and groove constant-velocity joint developed by Miller

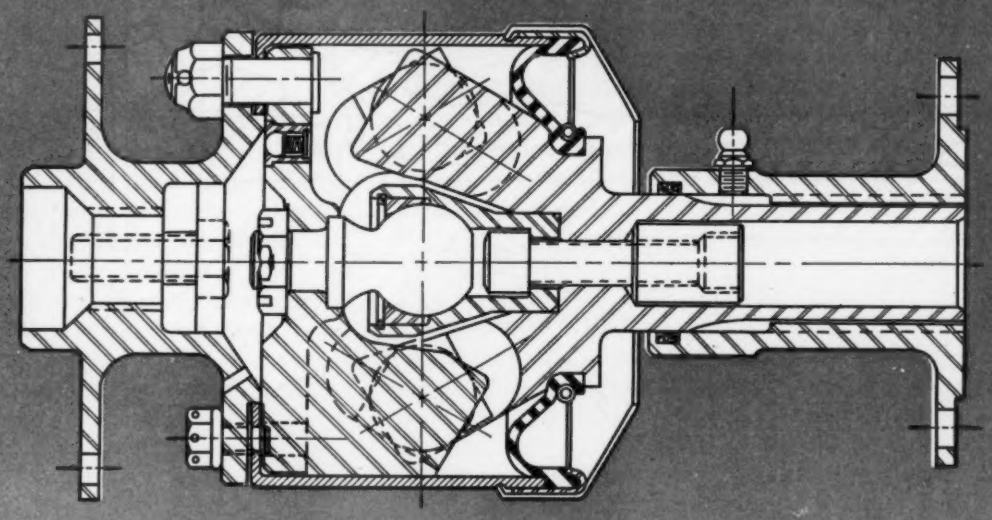
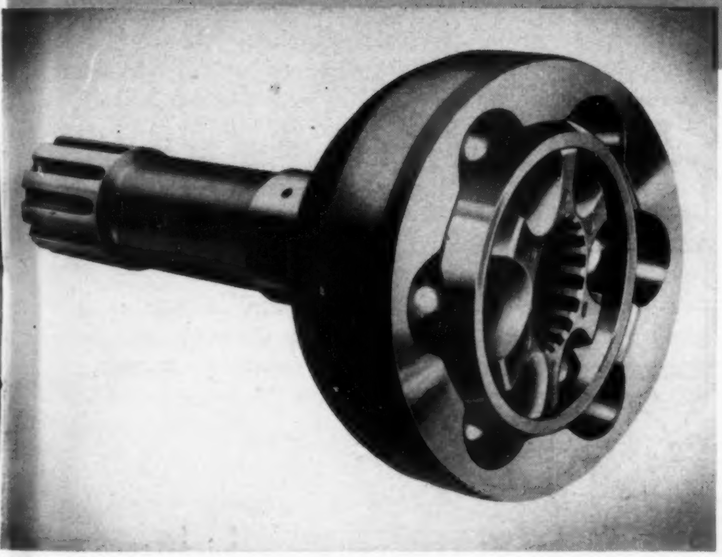


Fig. 8—Below—Rzeppa constant-velocity joint of the bell type showing ball and socket design with aligning cage



tained by means of a pin-shape lever having three bearings—in the inner race, outer race, and pilot—as shown in *Fig.* 9. Beyond an operating angle of about 11 degrees this device was not necessary, the crossing of grooves being sufficient for positive ball and cage positioning.

Today, the matching half ball-grooves are no longer formed from a common center but from two centers symmetrically located and offset an equal distance from the true joint center so that the half-grooves actually converge toward one side of the joint in a wedge fashion, compelling the six balls into the correct position by means of the ball cage. Sufficient offset makes self positioning of the balls so positive throughout the entire angular range of the joint that additional piloting parts are no longer required. Currently, however, pilots are still fitted in the largest joint sizes with nominal shaft diameters of 2¼ to 3 inches mainly for manufacturing reasons and as a safeguard against excessive stresses at operating angles from approximately 20 to 35 degrees.

Basic Types: Rzeppa joints are available in either disk or bell type in nominal sizes corresponding with their driveshaft diameters ranging from 15/16 to 3 inches. Disk and bell joints of the same nominal size have interchangeable internal parts. The distinguishing difference lies in the shape of the outer ball race and the shaft seal. Disk joints, *Fig.* 5, are furnished with a disklike, short cylindrical outer race having six holes for bolting the joint to a suitable companion flange on one side and to a cover on the other side. The cover is fitted with a flexible diphragm of oil-resistant synthetic rubber, through which the drive shaft extends, providing freedom to swing 18 degrees and to slide in or out. These joints are especially suited for high-speed propeller shaft drives where vibration is critical. The driveshaft rests, sliding or locked, in the splined inner ball race with its weight supported within the joint on spherical surfaces making external splined connections unnecessary and thereby eliminating misalignment, runout, and consequent vibrations.

Bell type joints are primarily designed as power steering drives for front-drive axles or articulated driving axles with independent wheel suspension. Here the outer driver is shaped spherically, *Fig.* 10, open on one side with the other side merging into a driving shank, which is usually part of the bell-shaped forging. These joints are capable of 37 degrees deflection. Lubricant seals are generally provided as part of the axle hub housing, but may be furnished separately as nonrevolving swivel housings surrounding the joint and having a flange for fastening to a suitable face of the wheel hub.

All joints have six driving balls fitted in hardened and ground grooves for transmitting torque simultaneously in either direction of rotation. A close fitting ball cage holds the balls in correct alignment. The joint assembly has considerable end-thrust capa-

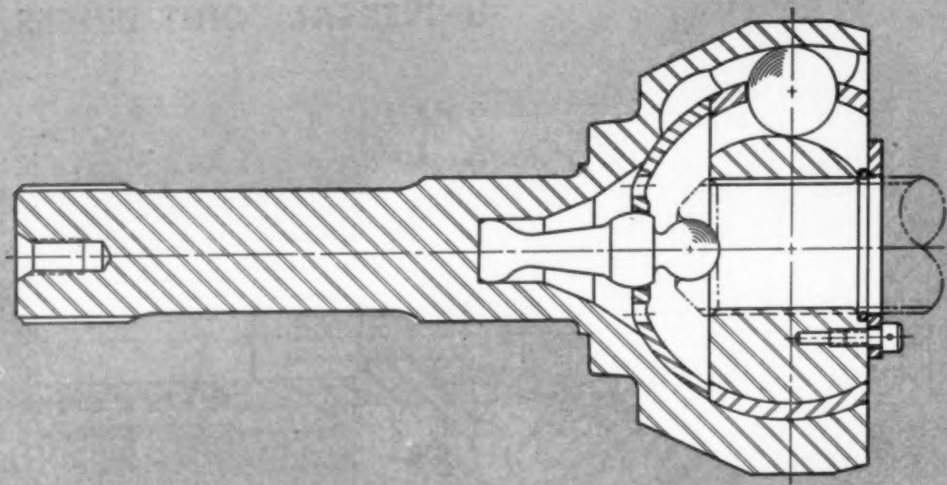


Fig. 9—Old style joint with pilot pin for locating cage at low angles

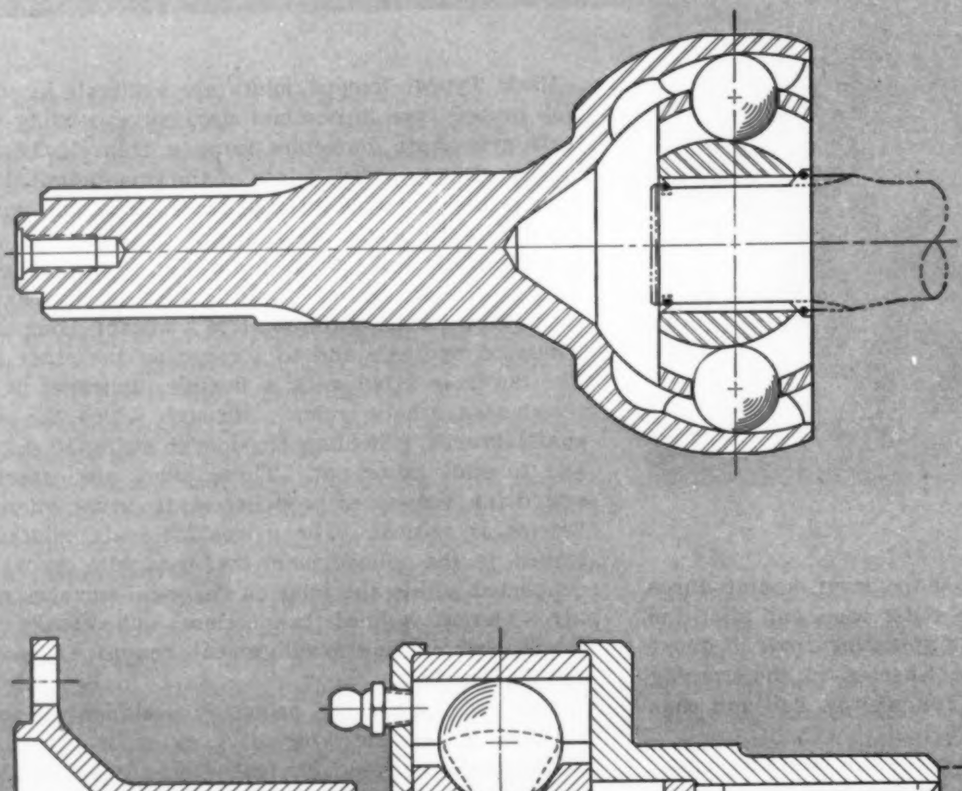


Fig. 10 — Bell type joint showing arrangement of parts

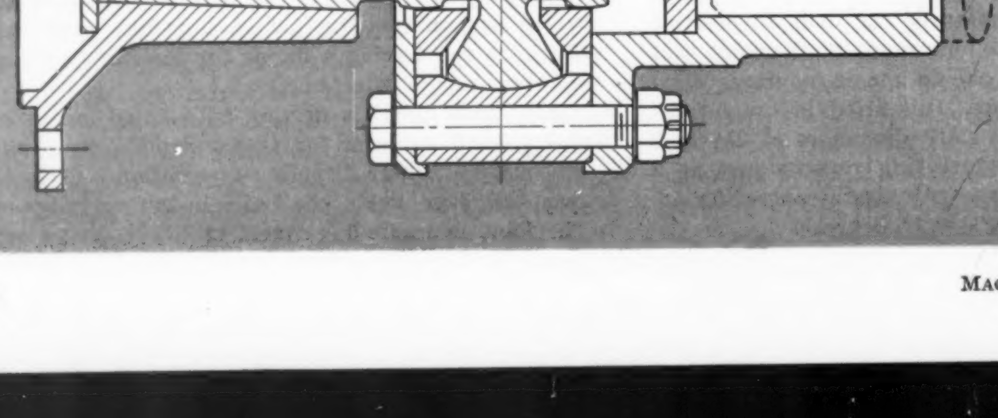


Fig. 11—Low-friction constant-velocity universal joint designed for moderate angles and high speeds

ity in either direction and no external supports are required to preserve the alignment of the individual joint parts. Only one shaft, either at the driven or driving end, requires support. The complete unit can be handled, mounted, and removed from an assembly without loss of internal alignment or possibility of accidental disassembly.

Joint Capacity: Obviously every universal joint has limited capacities for load, speed, angle, and the combination of these three factors. Load is generally governed by the permissible pressure in the journals of Cardan joints, or the flat surfaces of the Tracta, in relation to speed, angle and life expectancy.

For ball and groove joints, loads are determined by the size and number of balls and their distance from the joint axis. In accordance with ball-bearing practice, the permissible crushing load of balls will vary approximately with the square of the diameter. Since the ball distance to the axis of a joint is a multiple of its diameter, permissible load in congruent designs is proportionate to the cube of their linear dimensions.

Allowable ball pressures are experimentally determined with due regard to groove curvature, surface and contact conditions. With increasing joint angles contact conditions require a reduction of pressures in order to maintain the standard life expectancy under normal condition. Standard life expectancy is based on a fixed number of cycles which the joint will endure without excessive wear and is generally expressed in work-hours. It is identical for all sizes of joints under standard conditions of load, speed and angle. For any working condition different from standard the expected life can be predicted by the relation: $L = n^2(Cek/P)^3$ in which n designates the speed, rpm; C the normal rated load capacity, hp per 100 rpm; P the load, hp; and e and k, angle and speed factors. Centrifugal forces exerted by the balls on the groove surface at speeds far above normal will somewhat reduce load ratings (factor k).

Overheating due to internal friction or vibration caused by inadequate supports limit the speed or angle at which a joint may be safely run. As a rule, the product of speed in rpm and angle in degrees should not exceed a constant limiting figure set at approximately 16,000.

Efficiency: Universal joint efficiency is high; losses are chiefly due to internal friction and for some types are so slight they are not easily measured. Load tests with ball type joints operating at angles larger than 25 degrees at full load and low speed have shown losses of about 2 per cent, diminishing with the angle. There may be additional losses for Cardan joints operating under critical conditions where the variable speed will produce inertia effects detrimental to efficient power transmission. For instance, such variable speeds, necessarily resulting in variable torques, may produce stalling effects under critical conditions as may be encountered in front-drive vehicles negotiating steep, sharp turns requiring full steering lock.

Strictly speaking, these effects cannot be called mechanical losses but have practically the same effect. Nearly all losses are converted into heat and consequently sufficient cooling must be provided to avoid seizures of parts. At low speeds, such as are common in driving axles for vehicles, this condition is not critical and joints may be installed in closed housings

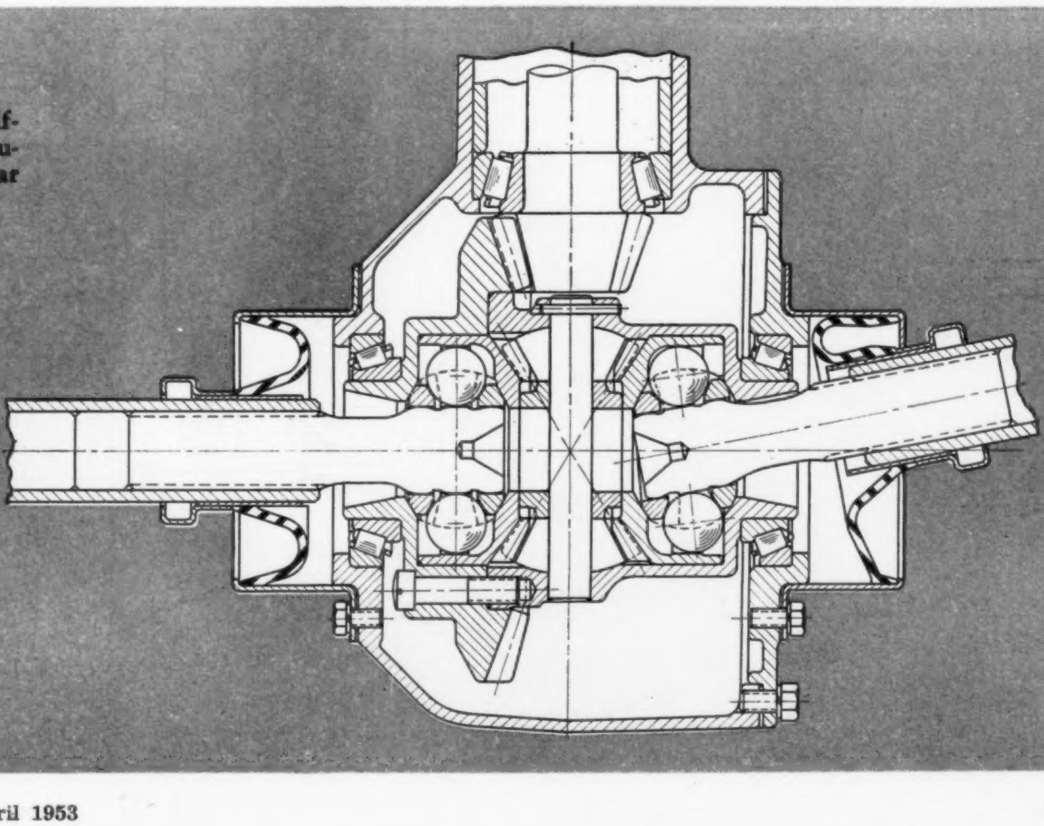


Fig. 12—Jointed differential for articulated front or rear axles

which also serve as reservoirs for lubricant. At high speeds, however, such housings usually hinder cooling of the joint, thereby reducing its capacity unless proper ventilation is provided.

Modern Developments: Recent innovations in ultra-high-speed engines, transmissions and shafts have brought newly developed types of universal joints capable of transmitting loads at small angles and higher speeds than was heretofore considered safe. A very successful joint of this type is shown in *Fig.* 11. This constant-velocity joint has a number of balls placed in straight half-grooves of the inner and outer joint race. Both races are interlocked by a spherical bearing which absorbs both radial and thrust loads. Here again the principle prevails that the driving balls must be located in a plane bisecting the shaft angles. This exact locating is controlled by two conical pilot rings seated spherically on the inner race shaft and guided by end enclosures of the joint. With the shafts at an angle, both pilots are displaced in opposite direction and locate the balls accurately between the conical surfaces.

Due to reduced internal friction and perfect balance, these joints can safely operate at speeds much higher

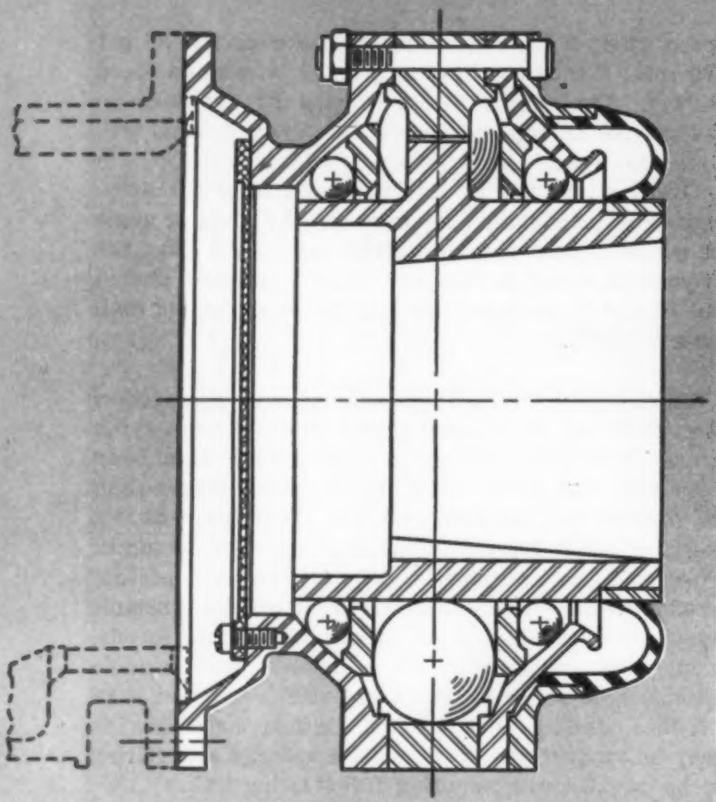


Fig. 13—Above—An extralightweight aircraft shaft coupling designed for 3 to 5 degrees deflection and ⅜-inch total lateral movement

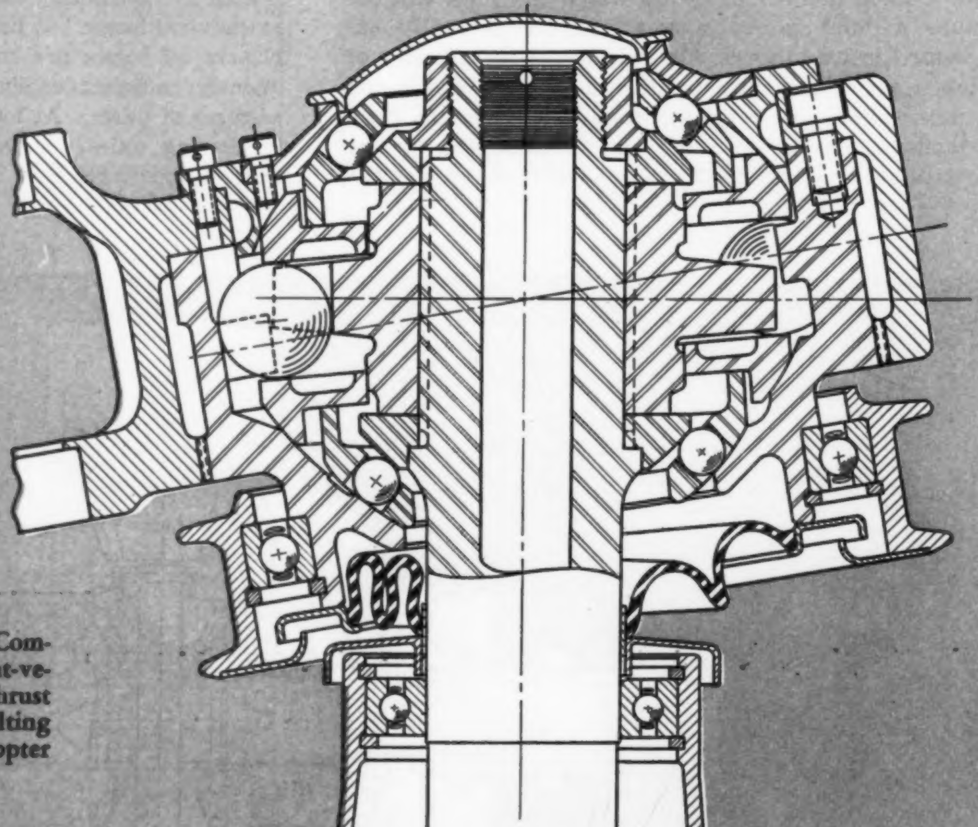


Fig. 14—Right—Combination constant-velocity joint and thrust bearing for the tilting rotor of a helicopter

than permissible for earlier types. Since in most high-speed installations the operating angles required are small, the joint is designed for a shaft clearance angle of 7 to 9 degrees. When larger angles are required at lesser speeds, the clearance can be incrased to approximately 12 degrees by certain modifications of the pilot guiding surfaces. In the combination of joint angle and speed the limits are set by vibration, heating and sealing of the lubricant but are much higher than the limit previously given.

Regardless of the method of suspension, the wheels of articulated driving axles are generally connected by jointed shafts to a differential housing centrally mounted on the spring-supported frame or body of the vehicle. In order to avoid excessive angles at full spring deflection it is important to provide for the greatest length of connecting shaft. This is particularly important for steering joints because the up or down movement due to spring deflection will deduct a certain amount of available steering angle. The compound angle may be roughly calculated as the square root of the sum of the squares of the component angles.

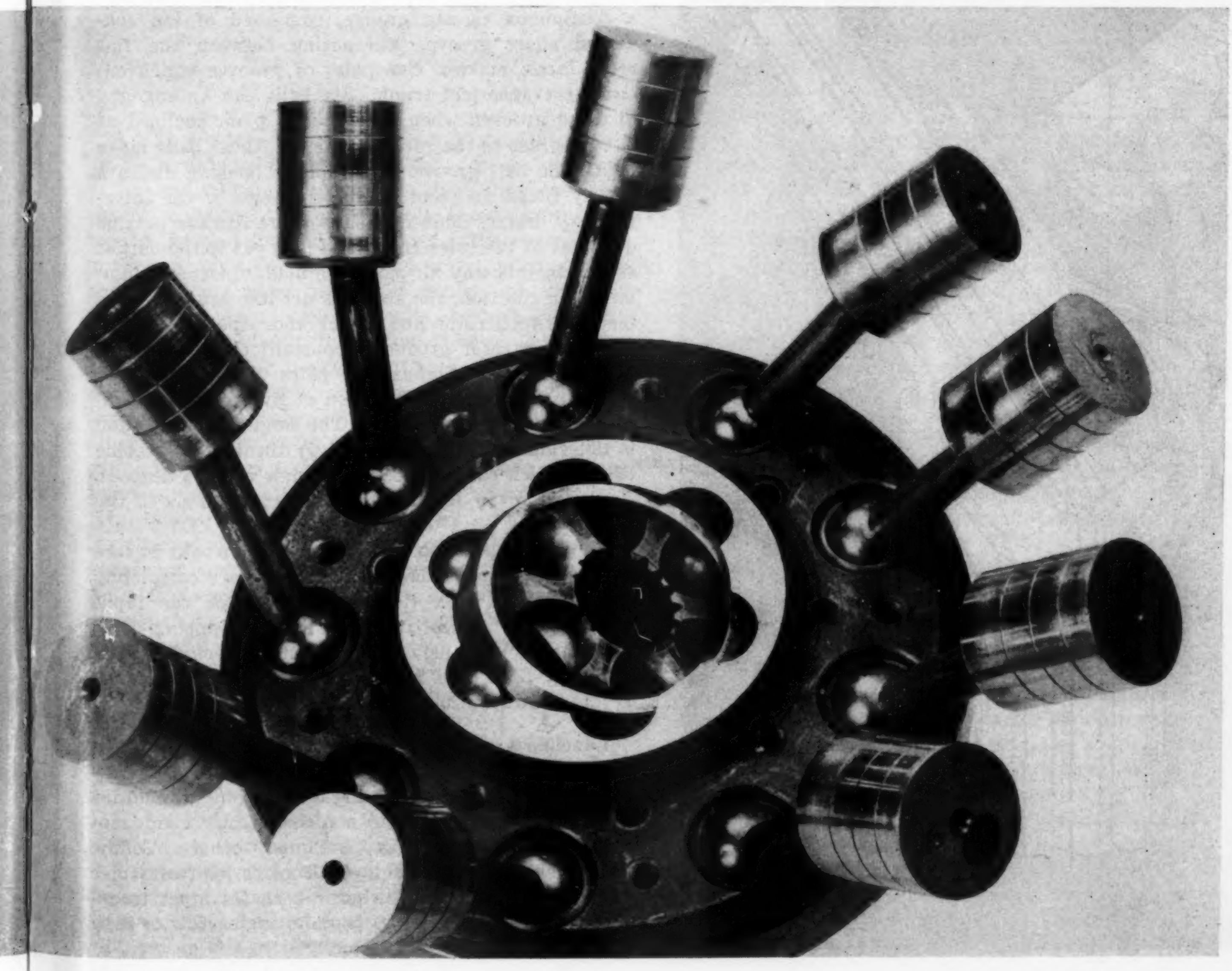
A solution for gaining maximum length for half axles is shown in ***Fig.*** 12. In this design the inboard

universals are developed as part of the differential side gears contained in the differential housing to form a very compact unit. For nonsteering rear axles, wheel joints may be mounted on the outside of the wheel hub, giving additional shaft length and easy accessibility.

Aside from light high-speed drive shafts, a number of special adaptations of constant-velocity universal joint drives have been built or designed especially for use in helicopters. In ***Fig.*** 13 is shown a light coupling capable of several degrees angularity as well as a small amount of axial movement to take care of deflections and length variations when mounted in the drive line to an outrigger rotor.

The function of this device can be plainly seen from the illustration. Axial movement of the inner race takes place on two ball bearings with a minimum amount of friction and without disturbing the action of the two disk-shape pilots which control the correct position of the ten large torque transmitting balls of

Fig. 15—Application of constant-velocity joint drive to the driving plate of a variable-displacement pump

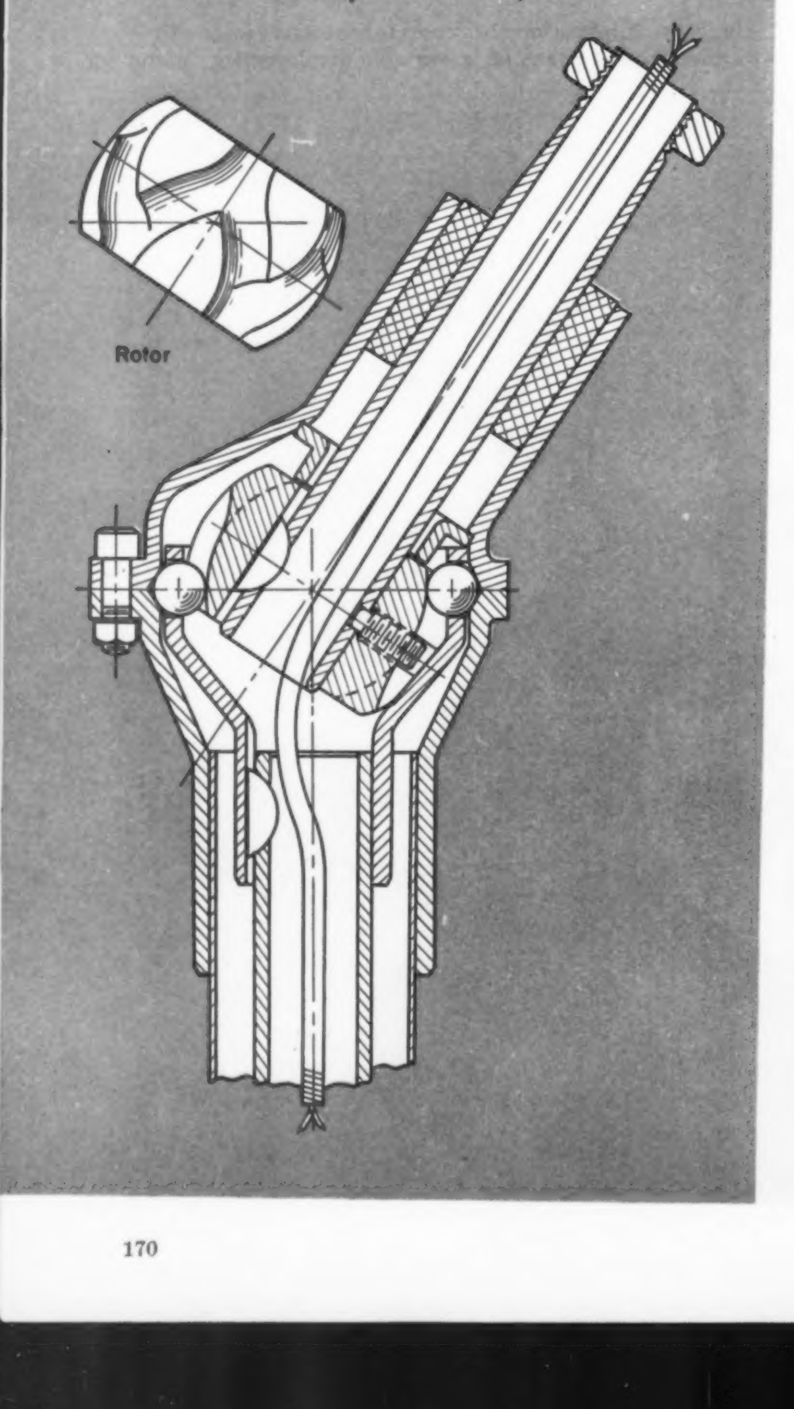


the universal joint. This coupling was designed for 400 hp at 2000 rpm and weighs approximately twelve pounds.

A tentative design for an autogyro rotor mounting is shown in *Fig.* 14. Here the vertical engine shaft terminates in a combination constant-velocity universal joint and thrust bearing, permitting the tilting of the rotor by an operator-controlled mechanism while the lift of the rotor is carried by the double-acting thrust ball bearing to the mounting on the pylon. Here again two ring-shape pilots control the plane of the ten driving balls which must bisect the shaft angle. They are correctly actuated by spherical surfaces whose centers are symetrically placed with respect to the true joint center on both shaft center lines.

Another unique application of a constant-velocity high-angle joint is shown in *Fig.* 15. The assembly is the driving plate for a series of plungers of a variable positive-displacement plunger pump. The plate, rotating sychronously with the multiple-cylinder block around the same axis but at a variable angle, is driven through the centrally located universal joint.

Fig. 16—Device for fixed-angle drive utilizing the principles of constant-velocity universal joint



Changing the angle of the plate with an external control mechanism changes the stroke of the pistons and pump delivery. Constant velocity here is of great importance because timing for the suction and discharge porting of the stationary cylinder head must be correct for all plate angles. A variable-speed joint would cause objectionable distortion of this timing.

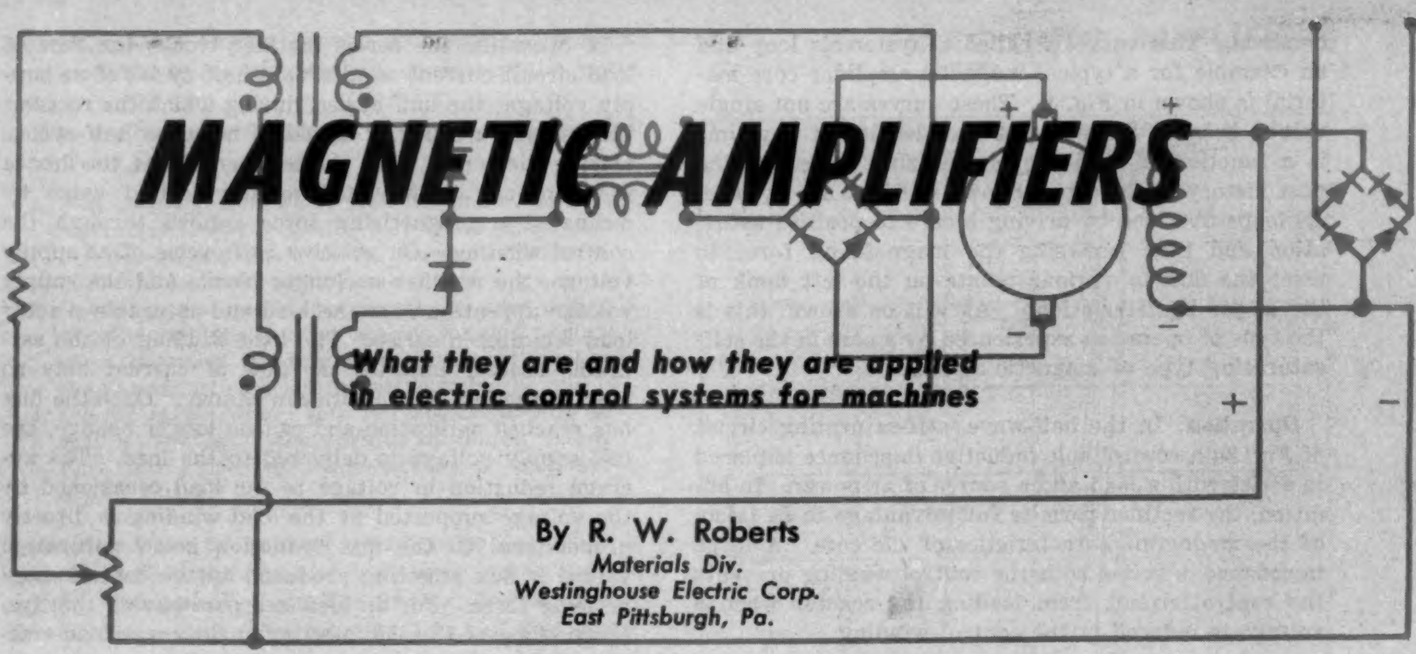
A torque transmitting angle drive developed by the author, although not a universal joint capable of variable angle, also may be grouped among the unusual applications owing to its features of constant velocity and torque transmitting balls. This device, *Fig.* 16, is designed for a fixed angle of 33½ degrees with a speed ratio of 1:1.2. In its present stage it is intended to transmit moderate torque at moderate speeds, such as are required for remote control mechanims, steering gears, machine tools, agricultural, road and construction machinery, marine and aircraft equipment, etc.

As shown in the illustration, the driving rotor has a continuous zig-zag groove, composed of ten connected short grooves alternating between the flat rotor faces, making five pairs of grooves which encircle the spherical trunk. Six balls can be engaged in these grooves when located in a plane inclined at 33½ degrees to the rotor. In action, these balls move in a split ball groove of the outer housing and are closely fitted at equal spacing in holes of the intermediary driver. This ring-shape part fits the outside diameter of the rotor freely and is keyed to the output shaft. In this way all parts are held in correct alignment. In rotation, the six balls act like semispherical teeth between rotor and driver and proceed through the endless rotor groove in constant contact.

Due to the fact that five pairs of grooves mesh with six balls, a speed relation of five to six is maintained between the shafts. The angle of inclination is determined by the ratio (5:6) through the cosine of 5/6 which corresponds to an angle of 33½ degrees. In this case the driver makes five turns to six of the rotor.

No thrust bearings are required as would be the case of bevel gears, because all forces are completely balanced with exception of one-fifth of the input torque which is absorbed in the housing support. This coupling is obviously of the constant-velocity type implying that both input and output shafts can be revolved at a uniform rate of speed.

Conclusion: The field of application for universal joints is extremely broad and encompasses a variety of drives, both low and high speed, in automotive vehicles, aircraft, ships, machine tools, road machinery, mining machines, and many others. Today, the modern counterpart of the Hooke's joint can solve critical drive problems wherever shafts must transmit power at variable or constant angles, low or high speeds, and at constant velocity.



MAGNETIC AMPLIFIERS

What they are and how they are applied in electric control systems for machines

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ANY electrical amplifier is basically a controllable impedance inserted between a source of power and a load. In useful amplifiers the power necessary to control this impedance is less than the controlled power delivered to the load. The controllable impedance may be resistive, inductive, or capacitive, and amplifiers have been built using each of these types. The prime example of a variable resistive-impedance amplifier is the vacuum tube while the magnetic amplifier, or magamp, utilizes a saturable reactor as a controllable inductive impedance. The saturable reactors, either alone or in combination with other circuit elements, provide amplification or control. Advantages and disadvantages of magamps are summarized in the accompanying listing.

In its simplest form, a magamp consists of a winding on a saturable ferromagnetic core placed in series between a load and a source of power. Since the controllable impedance is inductive and thus can only impede changes in current, the power source must be ac. A separate winding on the core controls the presaturation of the saturable magnetic core, thus controlling the impedance presented by the series winding between the load and ac power source.

Since the heart of a magamp is the saturable magnetic core, an understanding of the properties of magnetic cores is basic to an understanding of the operation of magamp circuits. For a winding on a magnetic core to develop a voltage that impedes the flow of current through the winding, there must be a change in the magnetic flux linking the turns of the winding. The time integral of voltage induced in the winding, or volt-seconds, is proportional to the product of the change in flux linking the turns of the winding, and the number of turns. In a saturable core in which the flux can change only between definite limits, there will be a definite maximum value of induced volt-seconds per turn.

The maximum ac voltage that can be applied to a winding on a saturable core without saturating the core is called the saturation voltage. At this value of voltage the volt-seconds supplied in each half-cycle produce the maximum possible flux excursion in the core. That is, with a value of ac voltage applied equal to the saturation voltage the flux will change from negative saturation to positive saturation value and then back to negative saturation on alternate half-cycles. The time integral of voltage, or volt-seconds, present in a half-cycle of ac voltage is inversely proportional to the frequency. Thus the maximum voltage that a winding on a saturable core can support will be directly proportional to the frequency, number of turns, and maximum flux change possible in the core.

Impedance of a winding on a magnetic core is determined by the ease with which the magnetic flux in the core can be changed. A reactor which requires a small change of current to produce a given flux change will have a higher impedance than one which requires a larger change of current to achieve the same flux change. The relation between magnetic flux and magnetizing current for a magnetic core material is usually plotted as flux density in gauss, or flux per square centimeter of core cross-sectional area, versus magnetizing force in ampere-turns per centimeter of magnetic path length times 0.4 π or

oersteds. This curve is called a hysteresis loop and an example for a typical magnetic amplifier core material is shown in *Fig.* 1. These curves are not single valued because the value of flux density at any time is a function of both the magnetizing force and the past history of the core. Shown in *Fig.* 2 are hysteresis loops obtained by driving a core to positive saturation and then reversing the magnetizing force to reset the flux to various points on the left flank of the major hysteresis loop. As will be shown, this is the type of operation experienced by a core in the self-saturating type of magnetic amplifier.

Operation: In the half-wave, self-saturating circuit of *Fig.* 3*a* a controllable inductive impedance is placed in series with a load and a source of ac power. In addition, the rectifier permits full advantage to be taken of the magnetic characteristics of the core. A large impedance in series with the control winding prevents the control circuit from loading the reactor when a voltage is induced in the control winding.

In operation the series rectifier blocks the flow of load circuit current on alternate half-cycles of ac supply voltage; the half-cycles, during which the rectifier blocks, will arbitrarily be called negative half-cycles. During the period when the rectifier blocks, the flux in the core can be adjusted to any desired value by means of a magnetizing force applied through the control winding. On positive half-cycles of ac supply voltage, the rectifier no longer blocks and the supply voltage appears across the load and saturable reactor load winding in series. The load winding of the saturable reactor impedes the flow of current only so long as the flux in the core can change. Once the flux has reached saturation and can no longer change, the full supply voltage is delivered to the load. The average reduction in voltage to the load occasioned by the voltage supported by the load winding is directly proportional to the flux reduction below saturation value, or flux resetting produced by the control magnetizing force. For an idealized core having the hysteresis loop of *Fig.* 3*b*, plotting the average load volt-

MAGAMP FEATURES

ADVANTAGES

Ruggedness—Magamps are static devices, extremely rugged and resistant to the effects of adverse environmental conditions. Similar in construction to transformers, they may be overloaded, like transformers, for extended periods with little effect except reduction in life. Life of a magamp control circuit is ordinarily determined by the life of associated selenium rectifiers which have life expectancies in excess of 20,000 hours.

Efficiency—Compared to vacuum tube amplifiers, magamps have high efficiency. No filament heating power is required and the only internal power losses, due to rectifier forward resistance and winding resistance, are small. Overall efficiency is usually better than 50 per cent and may be from 80 to 90 per cent in the larger sizes.

Amplification—A power gain of several million from a single stage amplifier is possible. However, since response time increases as amplification increases, more than one stage of amplification is usually used for high amplification to keep response time as low as possible.

Power Supply — Magamps operate directly from ac lines; they require no special power supply. By proper design, their operation can be made relatively independent of supply voltage variations. In contrast, vacuum tube amplifiers usually require large regulated dc power supplies.

DISADVANTAGES

Cost—Relative newness and low production rates make first cost high. However, cost of a magamp control system over a period of time may be lower due to savings in maintenance costs and fewer production stoppages because of breakdowns.

Frequency Response—Because magamps operate from ac supply voltages, they cannot be expected to respond to frequencies higher than the supply voltage frequency. Although amplifiers have been developed to give 100 per cent response within one cycle of control voltage, maximum frequency to be amplified should be maintained at approximately one-tenth of the supply frequency. An amplifier operated from a 60-cycle line should not be expected to amplify control signals with frequencies over 6 cycles per second.

Sensitivity—Dc input signals smaller than one microwatt cannot be satisfactorily handled by present commercially available magamps. Considerable effort is being made to improve sensitivity and it is expected that input powers as low as 10^{-18} watts may be satisfactorily handled with magamps.

Distortion—The output of a magamp is a highly distorted wave-form of the supply voltage frequency; the average value varies with the input control signal. Presence of the carrier frequency and harmonics is a severe handicap in certain applications.

age or current versus control magnetizing force in ampere-turns gives the control characteristic curve shown in *Fig. 3c*.

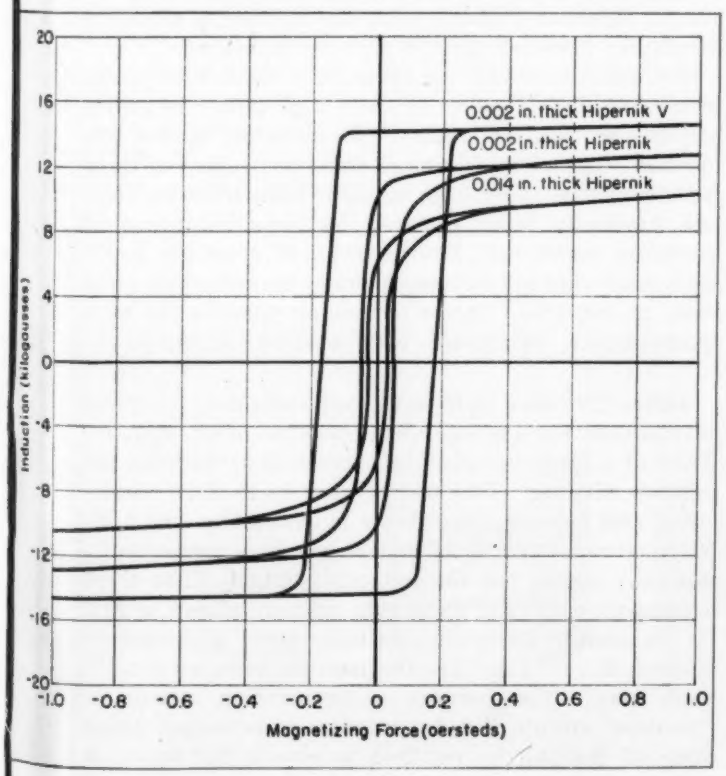
The control characteristic curve may be divided into three regions:

1. ***Full output region*** where flux remains at positive saturation during negative half-cycles of ac supply voltage and therefore the reactor cannot support voltage during positive half-cycles.
2. ***Control region*** where flux is partially reset during negative half-cycles of ac supply voltage, producing a reduction in average output proportional to the flux reduction below saturation value.
3. ***Cut-off region*** where flux is reset sufficiently so that it does not reach positive saturation at any time during the positive half-cycles of ac power supply voltage.

Amplification or gain of a magnetic amplifier is proportional to the slope of the curve in the control region. Because the reduction in average voltage delivered to the load is proportional to the flux resetting, the control characteristic curve of average load voltage versus control magnetizing force will be similar in shape to the left flank of the hysteresis loop for the magnetic core. Thus core materials with nearly vertical-sided hysteresis loops are required for high-gain magnetic amplifiers. If, in addition, a characteristic with a linear control region is desired, a core material with as rectangular as possible a hysteresis loop should be used.

Ac supply voltage for a magnetic amplifier is normally set at a value slightly below the saturation voltage of the load winding. Under this condition, to obtain cut-off, the flux must be reset nearly to the negative saturation value so as not to reach positive satur-

Fig. 1—Hysteresis loop of Hipernik core material



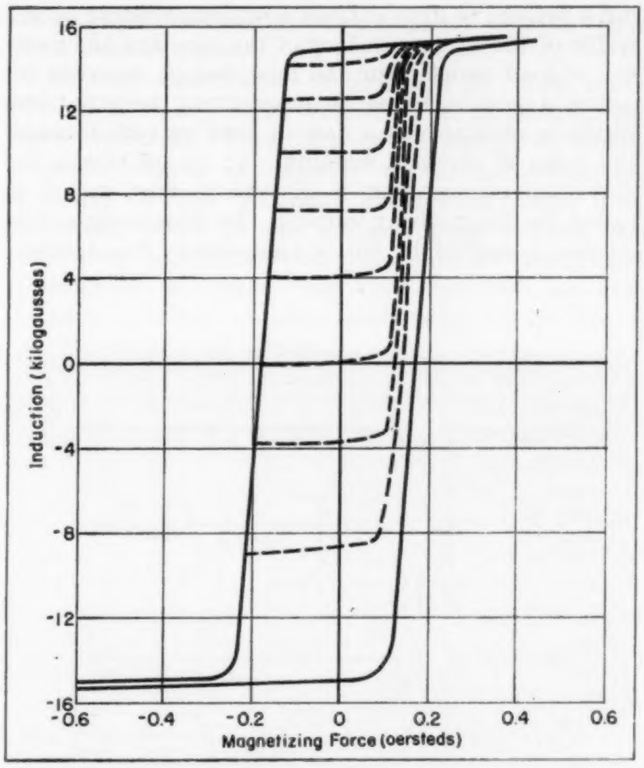
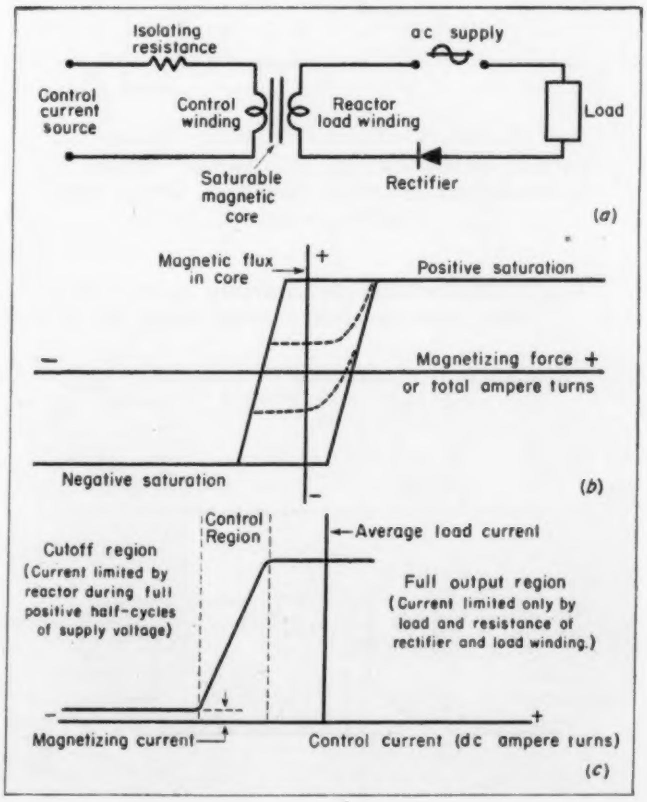


Fig. 2—Above—Hysteresis loops obtained by driving core to positive saturation and then reversing the magnetizing force to reset the flux to various points on the left flank of the major hysteresis loop

Fig. 3—Below—At *a*, basic half-wave self-saturating magamp circuit; *b* idealized characteristics of saturable magnetic core; *c*, control characteristics of half-wave magamp with idealized core



ation during positive half-cycles.

Load current that flows during the portion of the positive half-cycle, when the load winding is supporting a voltage to limit current flow, is dependent on the width of the hysteresis loop of the core and the number of load turns. For the flux change, the core requires a certain value of magnetizing ampere-turns which is obtained by a flow of load current through the turns of the load winding. At cut-off this is the only load current that is allowed to flow, and it is called the magnetizing current. In most designs this current is negligible, but in cases where it is troublesome, a core with an extremely narrow hysteresis loop is used.

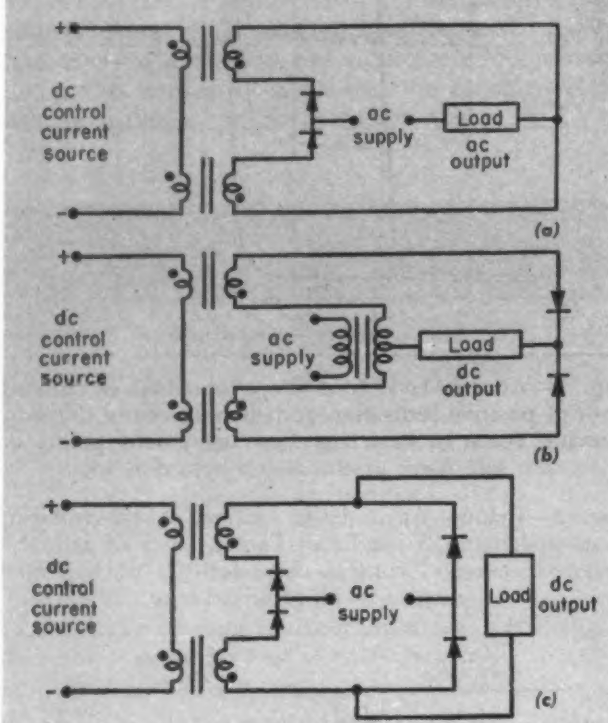
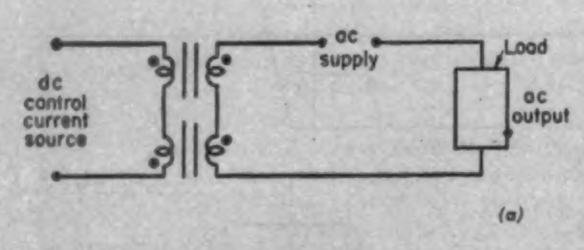


Fig. 4—Above—Typical magamp circuits: *a*, parallel self-saturating, or "doubler" circuit; *b*, center-tapped full-wave circuit; *c*, bridge type full-wave circuit

Fig. 5—Below—Simple saturable reactor magamp series circuit *a*, parallel circuit, *b*



Characteristics required of cores for use in self-saturating magnetic amplifiers can now be summarized. The major requirements are as follows:

1. The sides of the hysteresis loop should be as vertical as possible for high gain.
2. The width of the hysteresis loop should be narrow to minimize the magnetizing current.
3. The saturation flux density should be as great as possible to minimize the size of the core and the number of load turns required to support the ac supply voltage.
4. The hysteresis loop of the core material should be as rectangular as possible to produce a long, linear, and high-gain region of the control characteristic curve.

Characteristics required of a rectifier for use in a self-saturating circuit can now be analyzed. The forward resistance should be as low as possible to reduce losses; however, the reverse or leakage current is the most important characteristic to be considered. Any leakage current, during the period when the rectifier is supposedly blocking, will flow through the turns of the load winding and behave as an additional control current. The voltage that appears across the rectifier in the reverse direction and produces leakage is dependent upon the value of actual control current. This rectifier leakage current has the effect of negative feedback and reduces the gain of the amplifier. In addition to the reduction in gain produced by rectifier leakage, the variations in this leakage with ambient temperature are often particularly objectionable. Two approaches have been used in overcoming the difficulties introduced by variations in rectifier leakage. The best is to use low-leakage rectifiers and adjust the design so that this small leakage has negligible effect. In some applications the self-saturating rectifiers are shunted with a resistance to obtain the stabilization possible with the addition of negative feedback; however, gain is thereby sacrificed.

Selenium rectifiers are generally used with magnetic amplifiers because of their high power handling capabilities and reliability. Germanium diodes are used in some low-power amplifiers but because of their poor stability under high ambient temperatures, they are gradually being replaced by improved types of selenium rectifiers. Development of selenium rectifiers with very low reverse leakage characteristics has been an important factor in making possible the high performance obtainable with modern magamps.

Other Circuits: Half-wave, self-saturating magamp circuits are not generally used because of the requirement of a large isolating impedance in series with the control winding. One widely used method of eliminating this isolating impedance is to combine two half-wave circuits with their control windings connected to partially cancel out the induced voltages. The three commonly used circuits of this type are shown in *Fig. 4*. In each of these circuits the control windings are hooked in series so that the induced voltages tend to buck out. The parallel self-saturating circuit or "doubler" circuit, *Fig. 4a*, provides an ac output which may, if desired, be rectified to give a full-wave, dc

output. The "center-tap" full-wave, self-saturating circuit, *Fig. 4b,* provides a dc output directly; however, it requires a special center-tap transformer for the ac supply voltage. The bridge-connected, full-wave, self-saturating circuit, *Fig. 4c,* provides a dc output without the use of any additional parts other than the four rectifying elements.

Operation of these full-wave, self-saturating circuits is similar to the operation described for the half-wave circuit. When there is a high-impedance in series with the control windings, the operation is almost exactly the same; but with a low control signal source impedance, the operation changes since the voltages induced in the control windings do not exactly cancel. This change in operation results in a shift to the right of the control region of the control characteristic curve.

The choice of ac or dc output and the characteristics of the load are major factors in fixing the preferred type of circuit for a particular application. Another factor which affects the choice of circuit is the leakage characteristics of the available rectifiers. The reverse voltage that appears across the self-saturating rectifiers (the rectifiers in a series with the load windings) is different in each of the various circuits. The parallel or "doubler" circuit places the least reverse voltage across the self-saturating rectifiers, and for this reason it is often used in designs where rectifier leakage is troublesome.

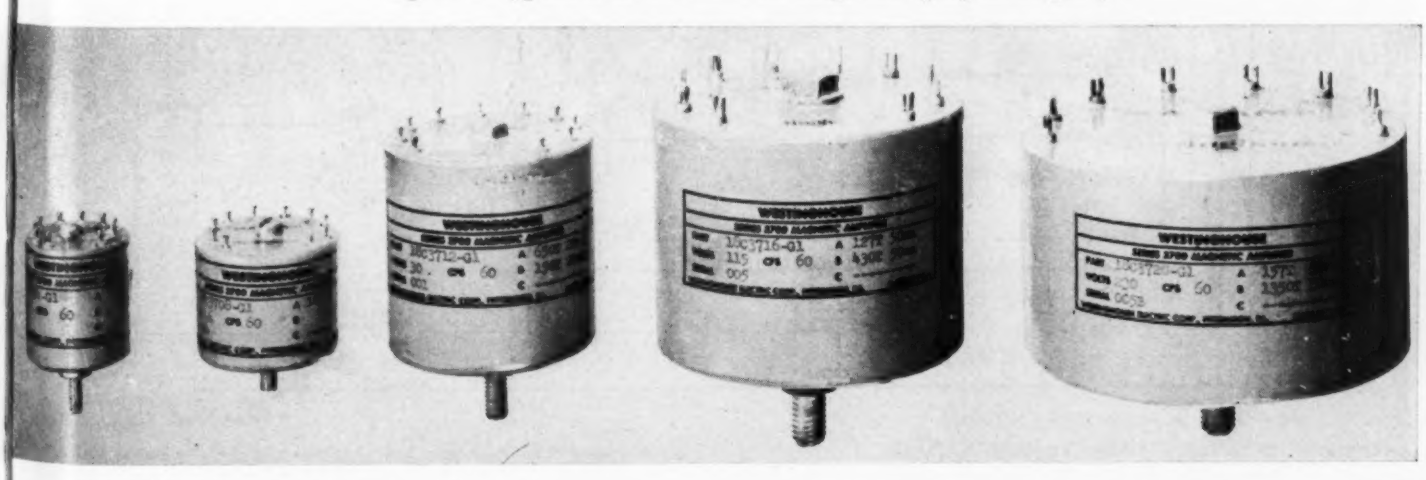
In addition to the various self-saturating circuits already discussed, many combinations of these basic circuits exist. Push-pull circuits have been devised for applications requiring a reversible polarity output. Two full-wave circuits are connected in opposition across a common load. Balancing the outputs of the two amplifiers gives zero voltage to the load. Then connecting the control windings of the two amplifiers so that one is driven up as the other is driven down provides a reversible polarity output. Another common application of the push-pull principle is in the use of magamps to drive reversible split-phase motors for use in servosystems. One method of accomplishing this is to connect two parallel self-saturating circuits, with ac outputs of opposite polarity, to the control phase of a two-phase ac motor. In other applications, usually those requiring higher power outputs, three-phase, self-saturating magamp circuits have been used extensively.

Another classification of magamps are those which do not have a rectifier connected in series with the load windings. These are called simple saturable-reactor circuits in contrast to the self-saturating circuits. Simple saturable reactor magamps were developed prior to the self-saturating circuits and were used quite extensively before the development of rectangular hysteresis loop core materials and low-leakage selenium rectifiers made practical the use of the self-saturating circuits.

Two basic types of simple saturable reactor magamp circuits are shown in *Fig. 5*. In these circuits the control windings are again connected in opposition so as to tend to cancel the induced voltages. The operation of these circuits is considerably different from that described for the self-saturating circuits. However, it is analogous to the operation of a self-saturating circuit with a large amount of negative feedback. The magnetic characteristics of the cores used in a simple saturable reactor circuit are much less apparent in the control characteristic of the amplifier than is the case in the self-saturating circuit. In the ideal case these amplifiers also require cores with narrow rectangular hysteresis loops; however, operation satisfactory for many applications can be obtained with poorer materials. In fact, practical simple saturable reactor magamps have been built using ordinary power transformer cores for saturable reactors.

The performance of simple saturable reactor magamp circuits is generally inferior to that obtainable with the self-saturating type of circuit. The ratio of power amplification to response time is much higher for the self-saturating circuits thus allowing a higher gain per stage with less time delay than can be obtained with simple saturable reactor circuits. The simple saturable reactor circuits do have certain other advantages such as stability and linearity which make them particularly useful in metering applications. They are being used extensively in the isolation metering of high dc currents. Current transductors, as

Fig. 6—A typical series of standardized general purpose magamps



they are called, are available with accuracies of one-fourth per cent for full scale currents of several thousand amperes dc.

Time Delays: In the operation of the half-wave, self-saturating magamp, there is a time delay in the response to a change in the control signal because a half-cycle of supply frequency must elapse before a change in flux resetting produces a change in output. The exact value of this time delay is dependent upon the particular point in the cycle at which the change in control current is made; however, in any case full response to any change will be obtained within 1½ cycles of the supply frequency. In the case of the full-wave circuits, in which the large isolating impedance in series with the control winding is no longer required, there can be a much longer time delay in the response of the amplifier to a change in control voltage. Here only a small dc control voltage is required to force control current through the resistance of the control windings. If a small change in this control voltage is made, it will take an appreciable length of time for the control current flowing through the inductance of the control winding to reach its new steady-state value.

Time delay can be reduced by placing additional resistance in a series with the control winding. This has the effect of requiring a greater control source voltage and thus making more voltage available for forcing a change in control current through the inductance of the control windings. However, additional power will be required to force the control current through this resistance, and thus the power amplification will be reduced.

The ratio of power amplification to response time in cycles has been found to be a constant for a given amplifier over a wide range of operating conditions. This ratio is called the figure of merit of an amplifier. In its calculation the power amplification is computed on the basis of the ratio of total power dissipated in the load circuit to the total power dissipated in the control circuit. The response time is generally expressed in the cycles of supply voltage necessary for a 63 per cent response in the output to a step change

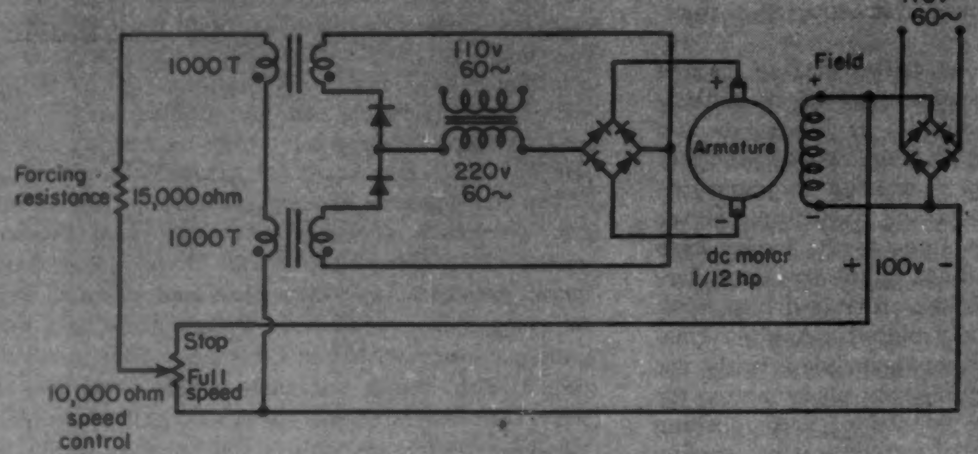
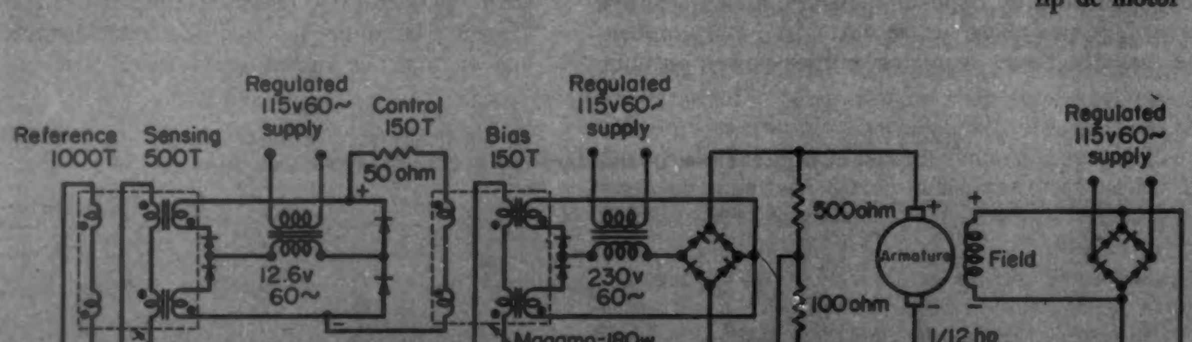


Fig. 7—Left — Magamp speed controller for 1/12-hp dc motor

Fig. 8—Below—Magamp speed regulator and control for 1/12-hp dc motor



of control voltage.

The figure of merit offers a convenient criterion for expressing the performance of an amplifier. For example, the figure of merit of a series-connected simple saturable reactor magamp is constant at a value of four per cycle. In comparison the figure of merit of high-performance, self-saturating magamps varies from about 200 per cycle to well over 5000 per cycle for amplifiers with higher output powers.

General Purpose Magamps: Some indication of the characteristics available with modern magnetic amplifiers is given by data for a series of Westinghouse magamps. This series is a set of high-performance magnetic amplifiers designed for general purpose applications in industrial and communications equipment control systems. The physical appearance and important operating characteristics of the amplifiers are shown in *Fig.* 6 and TABLE 1.

The operating characteristics of the series as tabulated in TABLE 1 are intended for operation in the parallel self-saturating or doubler circuit. The data is for ac output but a dc output can be obtained with the addition of a complementary bridge rectifier. The operating characteristics with dc output will be the same as for ac output except for the small losses in the added rectifiers.

The magnetic cores used in the series are Hipernik V, an iron-nickel alloy. Full advantage of the material's magnetic properties are realized by utilizing the essentially gapless, wound strip toroidal core construction. The resultant cores have a nearly rectangular hysteresis loop and a relatively low core loss. In addition, the toroidal configuration results in a smaller mean magnetic path length; thus fewer ampere-turns are required to establish a given magnetizing force than would otherwise be required. Toroidal construction also saves space and weight.

For each of the amplifiers of the illustrated series,

Table 1—Typical Series of Magamps

Characteristic	Nominal Power Output (watts)[1]				
	0.04	0.3	7.0	40	180
Nominal ampere-turns for control[2]	0.35	0.60	2.0	3.3	5.5
Ac supply voltage (v)	6.3	12.6	30	115	230
Frequency (cps)	60	60	60	60	60
Phase	1	1	1	1	1
Maximum usable incremental figure of merit[3]	170	310	410	1000	1200
Maximum incremental power amplification (3 cycle response)[4]	240	400	450	600	...
Output current[5]					
Rated (amp)	0.150	0.24	0.60	0.60	1.1
Max. (amp)	0.016	0.603	0.50	0.96	3.3
Weight of reactor (lb)	0.13	0.20	0.85	2.5	7.0
Reactor diameter (in.)	1¼	1½	2¼	3¼	4
Reactor height (in.)	1½	1½	2¼	2¾	2½

[1] Maximum obtainable within the linear range of the control characteristic curve for rated load.

[2] Control ampere-turns necessary to change the output from cutoff to nominal power output point.

[3] Ratio of maximum power amplification to response time.

[4] Amplification corresponding to three cycle response time over the linear range of the control characteristic curve for the optimum load resistance.

[5] Rms value if output is ac; average value if output is dc.

the core size, output windings, winding space available for control windings, and physical assembly are standardized. The complete operating characteristics, in a form independent of the number of control turns, and also information necessary for the design of the control windings are set down for each amplifier. Thus great flexibility is obtained by allowing the user complete specification of the control windings within the limitations of the available space and manufacturing techniques.

The reactors for the amplifiers of this typical series are vacuum cast in a thermosetting resin which assures a maximum of environmental protection and at the same time produces a relatively small and lightweight unit which is mechanically durable and easy to mount.

Typical Applications: Magamps have been applied in such diverse fields as servosystems for aircraft automatic pilots and voltage regulators for central station generators. They have been used to amplify the outputs of thermocouples, photocells, and ionization chambers as well as to control the speed of the largest steel mill drive motors. These are but a few of the many fields of application in which the use of a magnetic amplifier has resulted in a more satisfactory system than could be obtained with electronic tubes or rotating amplifiers.

The description of a few typical applications of magnetic amplifiers will serve to illustrate the potentialities of this new tool.

A basic example illustrating the application of magamps to a particular field is the speed control of a dc motor. There are many applications requiring a closely controlled variable-speed drive such as steel rolling mills, elevators, automatic machine positioning drives, drives for rotary cutting tools, etc. Dc motors have been widely used in applications requiring a variable-speed drive because they are relatively simply controlled by varying either the field current or armature voltage supply. Wide speed range applications usually require armature voltage control. Magamps may be used to supply a controlled voltage to the armature directly or, in the case of larger installations where a separate dc generator is used to supply a dc motor, a magamp may be used to control the generator field current. An example of this is a large tandem cold reduction mill drive. These mills, for turning out steel strip for cans, refrigerators, stoves, automobiles, and toys, consist of four or five stands in line. Strip thickness decreases as the metal travels through each stand and the speed of each successive stand increases accordingly. Each stand uses a dc motor rated at several thousand horsepower driven from a separate generator. A magamp regulator has been developed to control a four-stand tandem mill and has been found to equal or better the excellent performance obtained from the rotating-regulator systems now in use.

Wide-range speed variation of a fractional horsepower dc motor is another example of the control pro-

vided by magamps. For example, a standard 1/12-hp dc motor is to be controlled with a small 2-watt potentiometer. The maximum output power required from the amplifier is somewhat over 100 watts in order to supply motor losses and the 62 watts corresponding to the 1/12-hp rating as well as to provide for some forcing. The 180-watt nominal output magamp listed in TABLE 1 comes closest to meeting this power requirement. The circuit used is shown in *Fig.* 7. Output of the magnetic amplifier is rectified, with a bridge connected selenium rectifier, and fed directly to the armature of the dc motor. Ac supply voltage is standard 115 volt, 60 cycle and a step-up transformer is used to obtain the 230 volts for the magnetic amplifier. A separate bridge connected selenium rectifier is used directly from the 110-volt line to obtain field voltage for the motor and also as a source of control current for the magnetic amplifier. In order to keep the response time less than 0.1-second, the ratio of control turns squared to control circuit resistance, should be less than 1500. In the circuit shown in *Fig.* 7 a 1000-turn control winding is used, requiring a maximum of 5.5 milliamperes of control current. Approximately 100 volts dc will be available from the rectifier and with 15,000 ohms in series with the control winding, more than enough current will be available to obtain cutoff in the amplifier and stop the motor. A potentiometer of 10,000 ohms across the 100 volt dc supply will dissipate less than the allowable two watts. The ratio of control turns squared to control circuit resistance is always less than 70 and, from the data in TABLE 1, the response time of the amplifier will be less than four cycles of the 60 cycles per second supply frequency. Of course, the response of the complete system will be much longer due to inertia.

This system for controlling a motor with a small potentiometer is perhaps of little practical value by itself. However, if the small potentiometer were replaced by some sensing device that compared the motor speed to a reference and delivered an output proportional to the difference, then the amplifier could be used to regulate the motor speed. Such a system would be a feedback control system and could be made to give very accurate regulation of the motor speed at any desired value. Since feedback is present, care must be excercised in the design of such a system to avoid oscillatory behavior but, due to the excellent mathematical tools available for analyzing such problems, little design trouble is generally encountered.

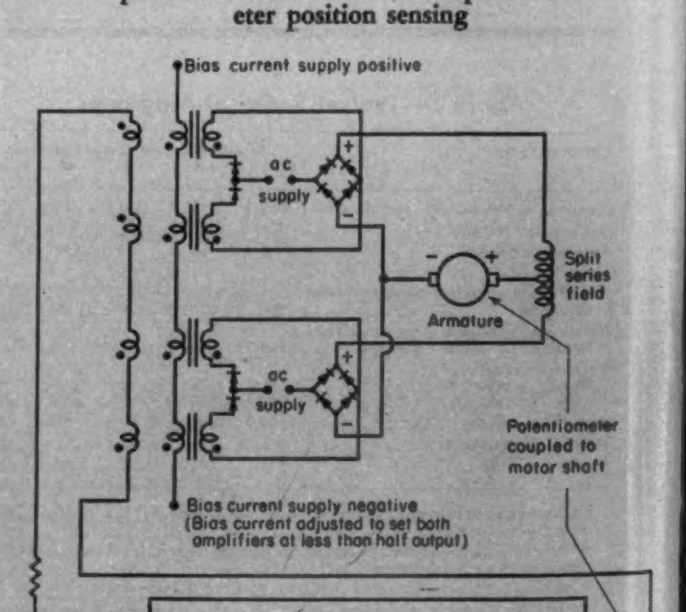
A feedback speed controller for the 1/12-hp dc motor is shown in *Fig.* 8. The speed sensing is obtained from the back induced voltage of the motor rather than from a separate tachometer in order to make the system self-contained and as inexpensive as possible. The back induced voltage of a motor is proportional to speed if the field current is constant; however, the resistance drop in the armature will modify this relation. A signal that is proportional to speed is obtained by sampling a combination of armature current and voltage. The reference for the system is the rectified and filtered output of a constant-voltage transformer. The constant output of the rectifier is also used for the motor field supply to make use of the speed-sensing scheme as shown. Filtering of the reference signal and a speed-sensing signal is necessary because each has a different wave form and can only be compared on the basis of their average values.

Two stages of amplification are required with the feedback control system in order to obtain sufficient amplification to achieve the desired accuracy and yet not introduce excessive time delays. Comparison of the speed-sensing signal and reference signal is accomplished magnetically in the first stage magamp. The full reference current alone would drive the first stage far into the full output region while the sensing current at full motor speed would drive the first stage far into the cutoff region. The two signals at the same time will result, under equilibrium conditions, in the first stage operating somewhere in the control region. The second stage has a bias winding in addition to a control winding connected to the output of the first stage. With no output from the first stage the bias will place the second stage in the cutoff region. Full output from the first stage will drive the second stage into the full output region. Thus in operation, a reduction in motor speed below the equilibrium value set by the reference potentiometer will cause the speed sensing signal to drop, which increases the first stage output. The increased control to the second stage will increase its output causing the motor speed to increase.

The system as shown gave speed control over a ten to one range with regulation from no load to full load of five per cent at full speed and of eight per cent at 25 per cent of full speed. The response time for a step change in reference signal was 1.6 seconds with a heavy flywheel on the motor.

Position Servosystems: An increasingly important role is being played by position servos in the field of

Fig. 9 — Magamp servo amplifier using split series field dc motor and potentiometer position sensing

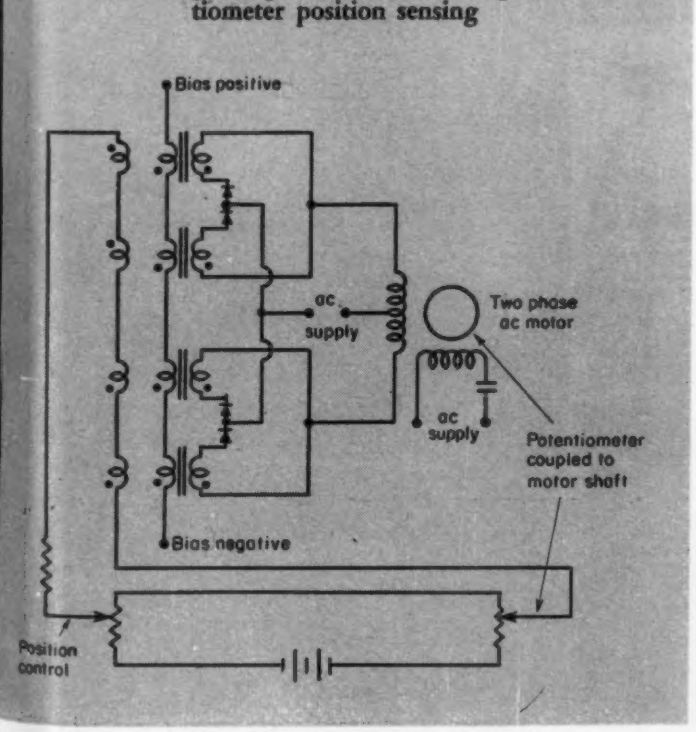


machine design. The era of the truly automatic machine is dawning and the muscles and nerves of the robot are position servos. They were used extensively during World War II for gun training, radar indicators, aircraft control systems, and in numerous other applications. Many of these existing applications are being re-evaluated at the present time from the standpoint of replacing vacuum tube and relay amplifiers with magamps.

A position servosystem consists basically of a device to be moved, a motor to move it, a sensing element to indicate position, a reference with which to compare the sensing signal, and a power amplifier to amplify the error signal and operate the motor. On a machine tool such as an automatic milling machine, the device to be moved might be the work-holding table. The drive might be a split-phase induction motor that moved the table through a set of reducing gears, a lead screw, and a drive nut. The sensing signal could be a potentiometer operated directly from the table by a rack and pinion gear drive. The reference signal could be either an angular position on another potentiometer or a corresponding voltage from a master computer. The amplifier would be magnetic and probably consist of two stages.

Drive motors for magnetic amplifier position servosystems may be either reversible dc or two-phase ac motors. Each has certain advantages, and a choice between the two types must be based on the requirements of a particular application. In general, a system using a dc motor is the smaller and more efficient, but reliability is limited by mechanical brush wear. A two-phase ac induction motor can be made with good mechanical reliability and, since there are no brushes, lower friction losses are possible.

Fig. 10 — Magamp servo amplifier circuit using two-phase ac motor and potentiometer position sensing



A few examples demonstrate the manner in which a magnetic amplifier is used in a position servo. A simplified circuit using a dc motor with a split-series field is shown in *Fig.* 9. The variations of possible amplifier circuits using dc motors are numerous. Small servos frequently use dc series motors as shown, while intermediate or larger systems often use shunt motors. Potentiometers are used as position sensing elements, although with slightly more complex circuitry synchros could be used to sense position. The left-hand control is used to set the desired position. The right-hand potentiometer is mechanically connected to the device that is to be controlled. Any difference in the settings of the positions of the two potentiometers produces an error signal in the amplifier control windings. This signal causes the amplifier to drive the motor in the proper direction to correct the error. The magamp consists of two separate amplifiers connected so that each alone would cause the motor to rotate at full speed in one direction or the other. A bias current, applied to each amplifier, operates both in the linear control region with equal outputs. Under this condition, there is no net dc delivered to the motor and it remains stationary. The control windings are connected so that a control signal increases the output of one amplifier and decreases the output of the other, thus causing the motor to rotate in a direction dependent on the polarity of the control signal.

A position servo using a two-phase ac induction motor is shown in *Fig.* 10. The system of control is identical to that used for the dc motor servo. Reversal of the ac motor can be obtained by changing the phase of the voltage applied to the motor control phase by 180 degrees. The two amplifiers are connected so that the voltages delivered to the motor by each are 180 degrees out of phase. Thus, if each amplifier has equal output, the net voltage to the motor will be zero and it will remain stationary. A control signal will cause the output of one to increase and the other to decrease, thus applying a voltage to the motor of a phase dependent upon the polarity of the control signal.

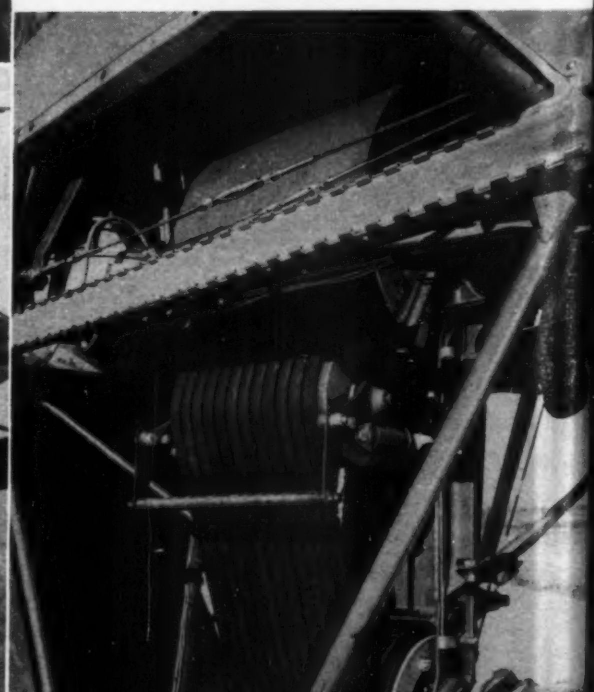
Important operating characteristics of a position servo are accuracy, response time, and stability. Accuracy is dependent upon the system gain. Two or more stages of amplification are generally required for a practical application. Response time of a system is dependent on the natural frequency of the motor and its load as well as the time delays introduced by the amplifier. The overall response time of a system is normally as short as possible so that controlled devices follow input control changes with little error. The problem of obtaining system stability with the required accuracy and response time is one of the more difficult phases of servo design. Great strides are being made in this field, and mathematical tools are available which enable a designer to determine the combinations of amplifications and response times which will produce the desired result without any undesirable tendency toward oscillation.

CONTE

Tandem Helicopter Has

FREEDOM from torsional vibration, which places excessive strain on rotor blades, is one of the features of the MC-4 helicopter, *Figs.* 1 and 2. Rotor pitch control of the McCulloch Motors Corp. helicopter, although conventional, emphasizes simplicity and complete interchangeability of front and rear rotor parts.

Both rotors are powered from a horizontal 3-inch diameter aluminum drive-shaft running the length of the helicopter, *Fig.* 3, which transmits power to each rotor through a right-angle reduction unit. All components of this system are interconnected through self-aligning couplings. A V-belt central drive, *Fig.* 4, links the engine to the rotor transmission system. Twelve V-belts transmit power from the lower pulley to the upper pulley, which contains a freewheeling unit to permit autorotation of the

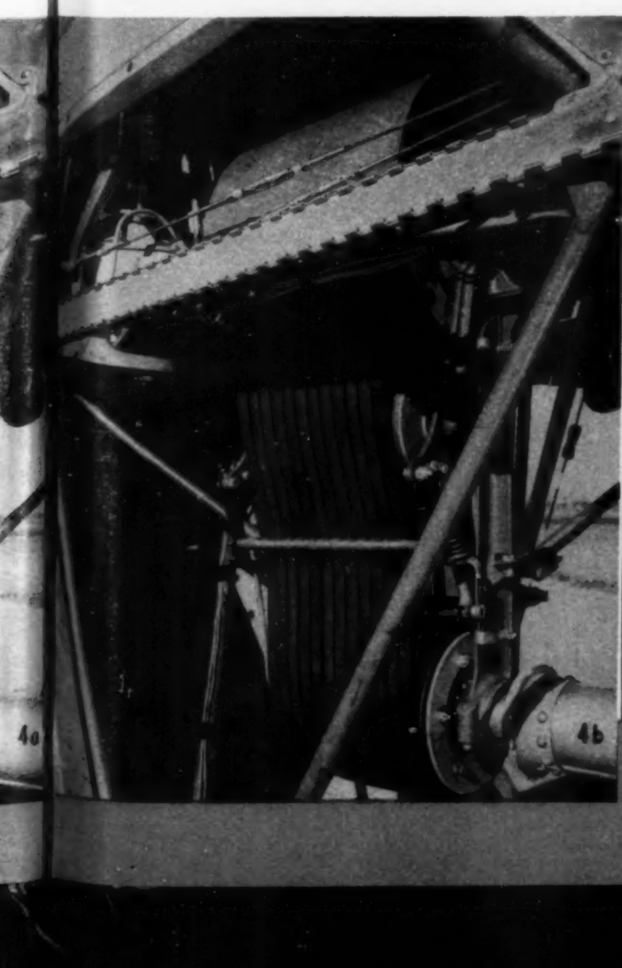


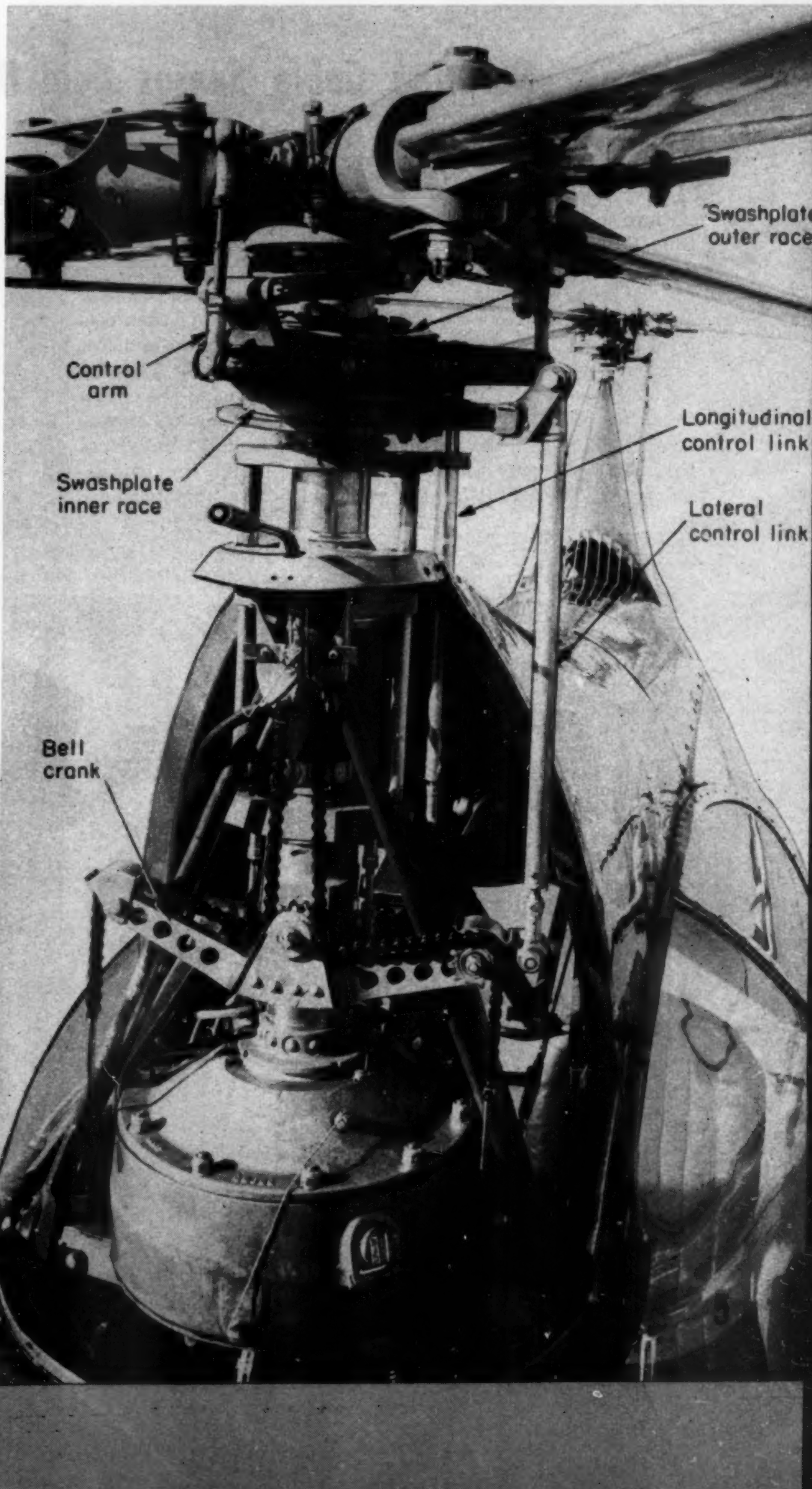
EMPORARY DESIGN

as Unique Transmission

rotors in case of a power-plant failure. Clutch operation is by means of a idler pulley controlled from the cockpit, shown engaged in *Fig. 4a* and disengaged in 4*b*.

Rotors are of the fully articulated type, with changes in rotor-blade pitch accomplished by a swashplate mechanism. Depressing a rudder pedal, for example, tilts the swashplate laterally, *Fig.* 5, through a chain-and-cable control system which turns a bellcrank linked to the stationary inner race of the swashplate. The swashplate outer race, which turns with the rotor assembly, is provided with three control arms connected directly to the blade pitch-control horns. Turning is accomplished by inclining rotor swashplates in opposite directions; lateral (sidewise) movement by inclining both in the same direction; and longitudinal





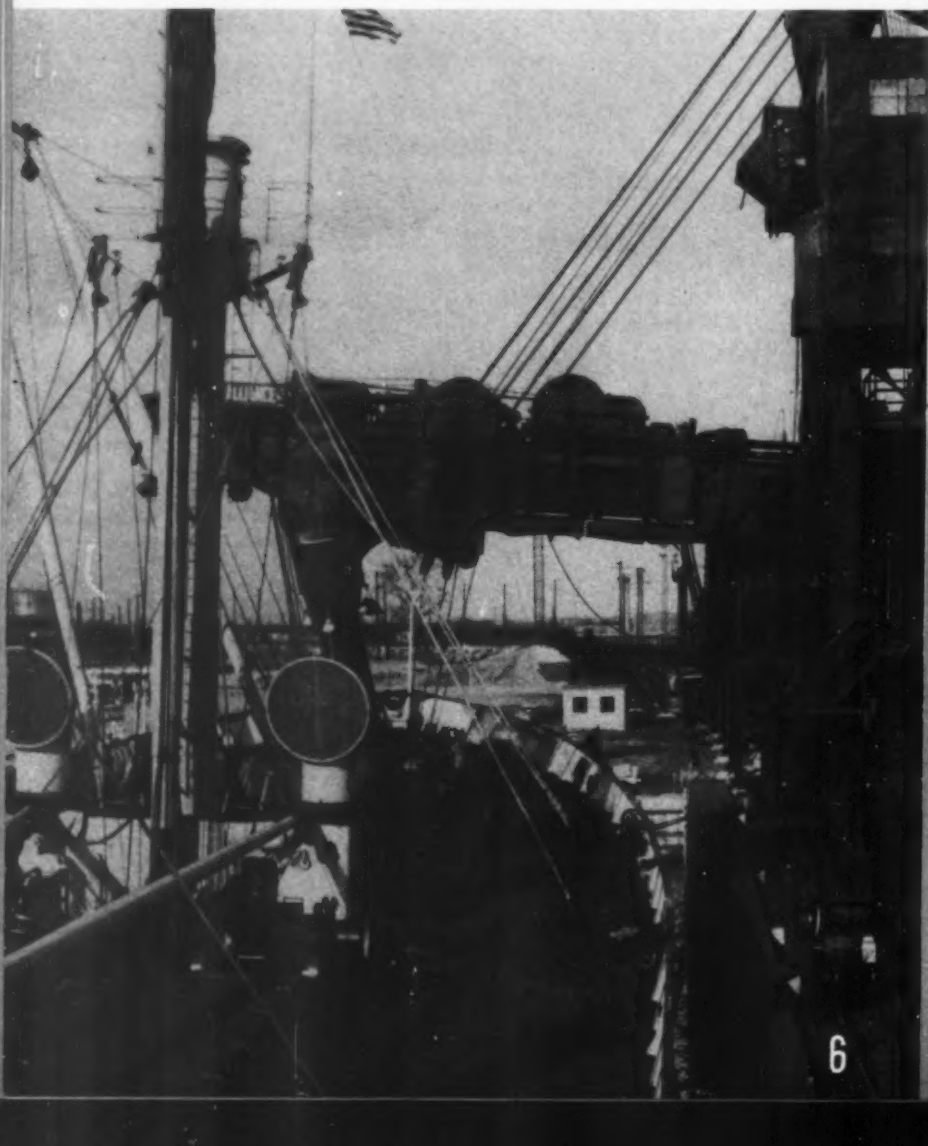
(forward and backward) movement with a push-pull system operating the longitudinal control link to tilt the swashplate.

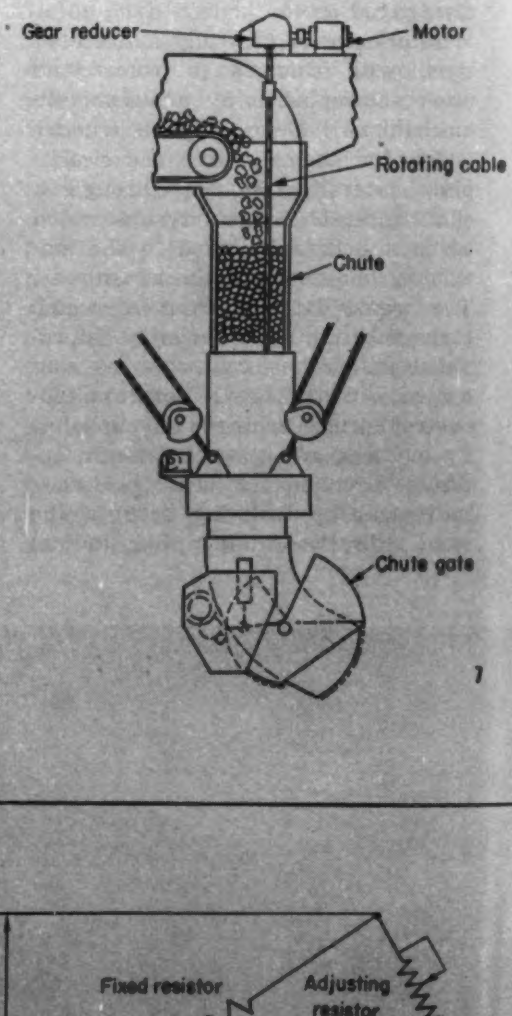
Additionally, collective pitch of all blades can be adjusted by moving the entire swashplate vertically, since the whole mechanism is sleeve-mounted on the drive shaft. Thus, upward or downward movement of the helicopter is controlled by a separate pitch-control lever, which also controls the throttle to compensate for the change in horsepower requirements.

Rotating Cable Senses Coal Height in Loader

TO PREVENT breakup and degradation of coal dropped into a ship loader, *Fig.* 6, J. A. Kell of Chesapeake and Ohio Railway Co. has developed a "servomechanism" for automatic control of coal height.

To sense coal height, a wire cable hanging inside the loader, *Fig.* 7, is rotated by a motor and gear reducer. When coal level is high, torque resistance on the cable causes the motor to slow down, unbalancing a bridge circuit, *Fig.* 8. This unbalance causes the chute gate to open, rebalancing the circuit by means of the repeat-back device on the chute gate. As coal level starts to fall, the motor speeds up, causing an unbalance that closes the gate to a point where the bridge circuit is rebalanced. Thus, coal level is maintained at a predetermined level.





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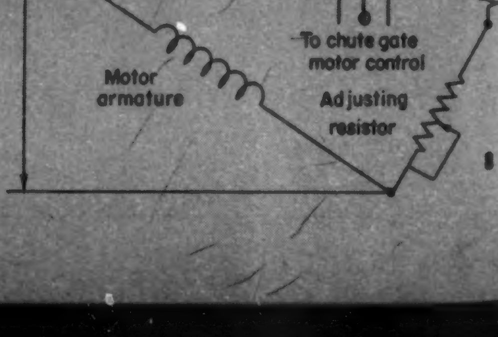

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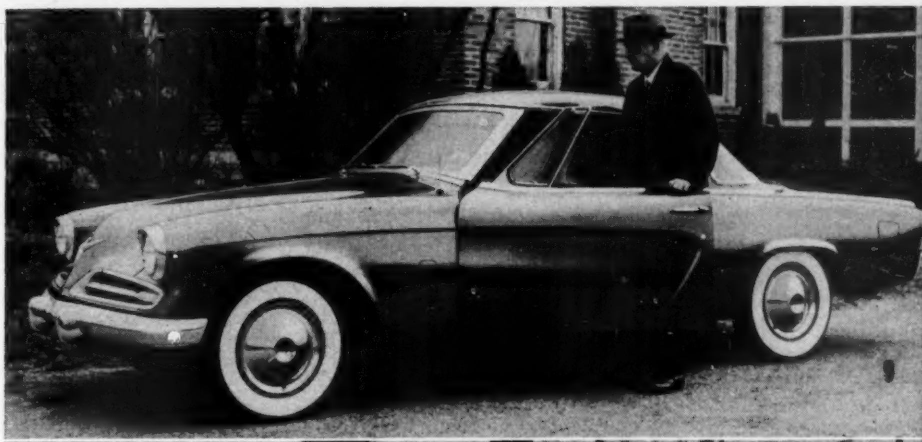
Studebaker Features Mechanical Power Steering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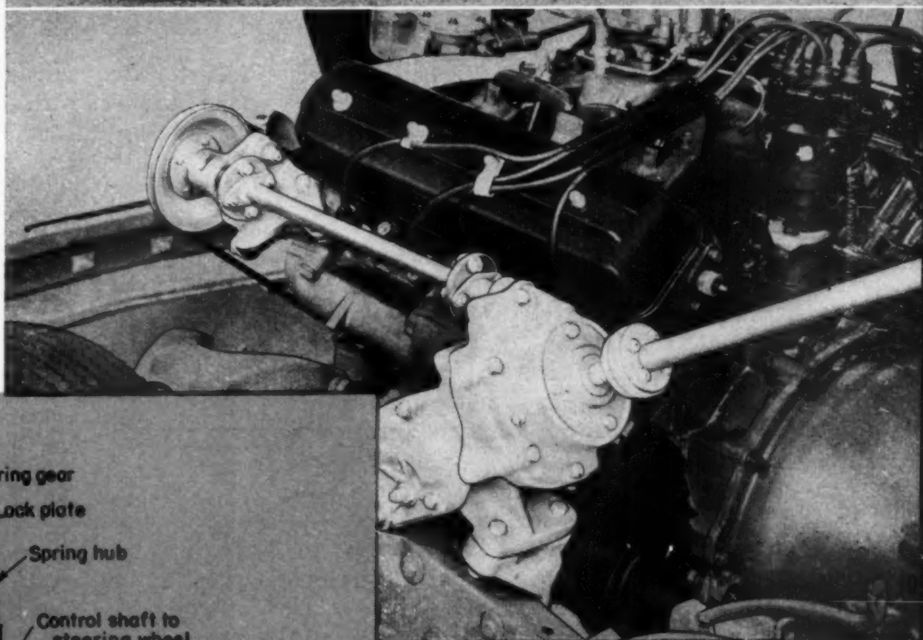
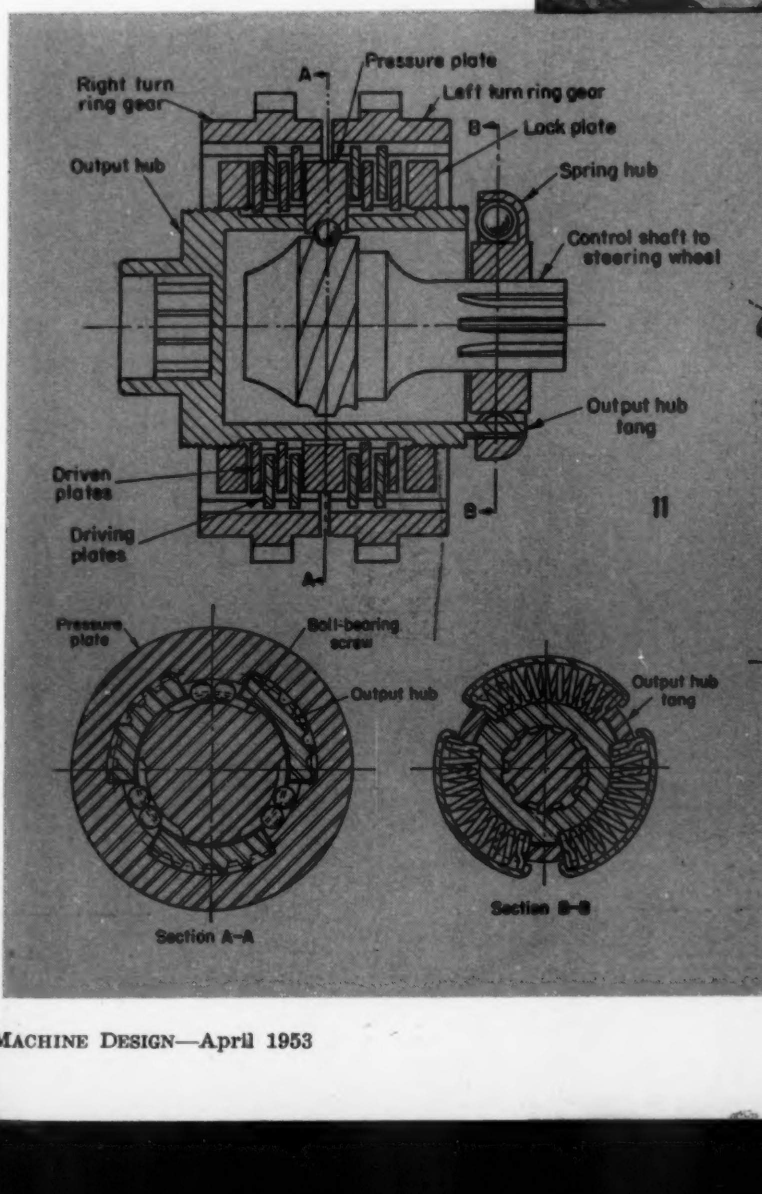


TWIN FEATURES of the 1953 Studebaker, *Fig.* 9, are the Continental styling (only 56¼ inches ground-to-roof height) and the new all-mechanical power-steering unit, *Fig.* 10. Power for the unit, less than 0.12-horsepower at 45 mph compared with about 0.94-horsepower for a hydraulic power-steering unit, is delivered by a belt-driven input shaft.

Two counter-rotating clutches geared to this shaft, *Figs.* 11 and 12, are the heart of the unit developed by Warner Gear Div. of Borg-Warner Corp. Each consists of a ring gear splined to the driving plates, with driven plates splined to the output hub. When the steering wheel is turned, a ball-bearing screw on the control shaft causes the pressure plate to move axially, applying engine power—at a 5 to 1 reduction in speed—to the output hub. As the output hub rotates, it rotates the pressure plate to the indexed position



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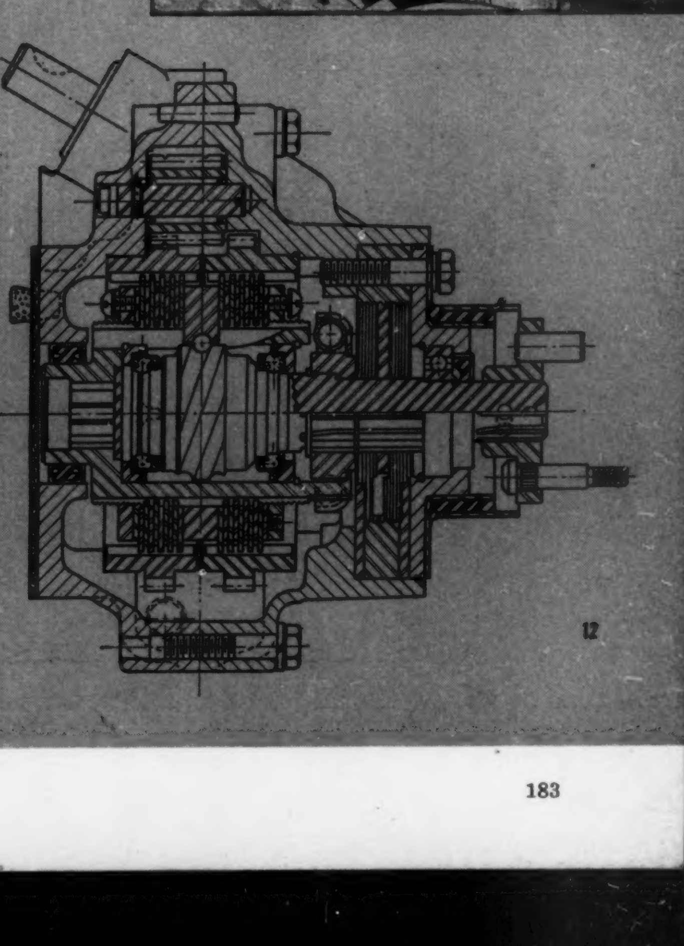
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(see section *A-A*), thus releasing the clutch.

A "manual feel" is imparted to the action by the spring hub (section *B-B*) splined to the control shaft. Three tangs on the output hub are centered by springs which can be replaced to adjust the "feel." When steering effort required is small, the spring hub turns the output hub directly because of the centering action of these springs. If required torque rises above the spring preload, relative rotation between the spring hub and output hub occurs, and the power steering unit is brought into play. When the steering wheel is released, the unit imposes no interference with natural recovery forces set up by the steering geometry of the car. If power fails, a ratchet on the input shaft permits the mechanism to overrun, and the car steers manually through the spring hub.

Constant-Force Spring Meters Level-Measuring Cable

CABLE is metered under constant force through 35 feet of travel by a neg'ator spring in the Levelrator level-measuring instrument, *Fig*. 13. As float position changes in accordance with liquid level, motion is transmitted by the cable to a drum, *Fig*. 14. A constant torque of 1.1 lb-in., delivered by the neg'ator regardless of float position, counterbalances the float without cable slippage or unwinding if sudden float movements occur.

Replacing multiple-spring systems, which are unsatisfactory for constant force over large distances, and bulky dead weights, the constant-force spring "motor" also drives a dial and counter. Cable traverse across the face of the cable drum is by means of a screw-driven guide operated by chain and sprocket from the drum output shaft, *Fig*. 15. The Fischer and Porter Co. instrument has an accuracy of $\pm\frac{1}{8}$-inch, and can be furnished with electric, pneumatic or selsyn transmitting units driven by the same system for remote indication.

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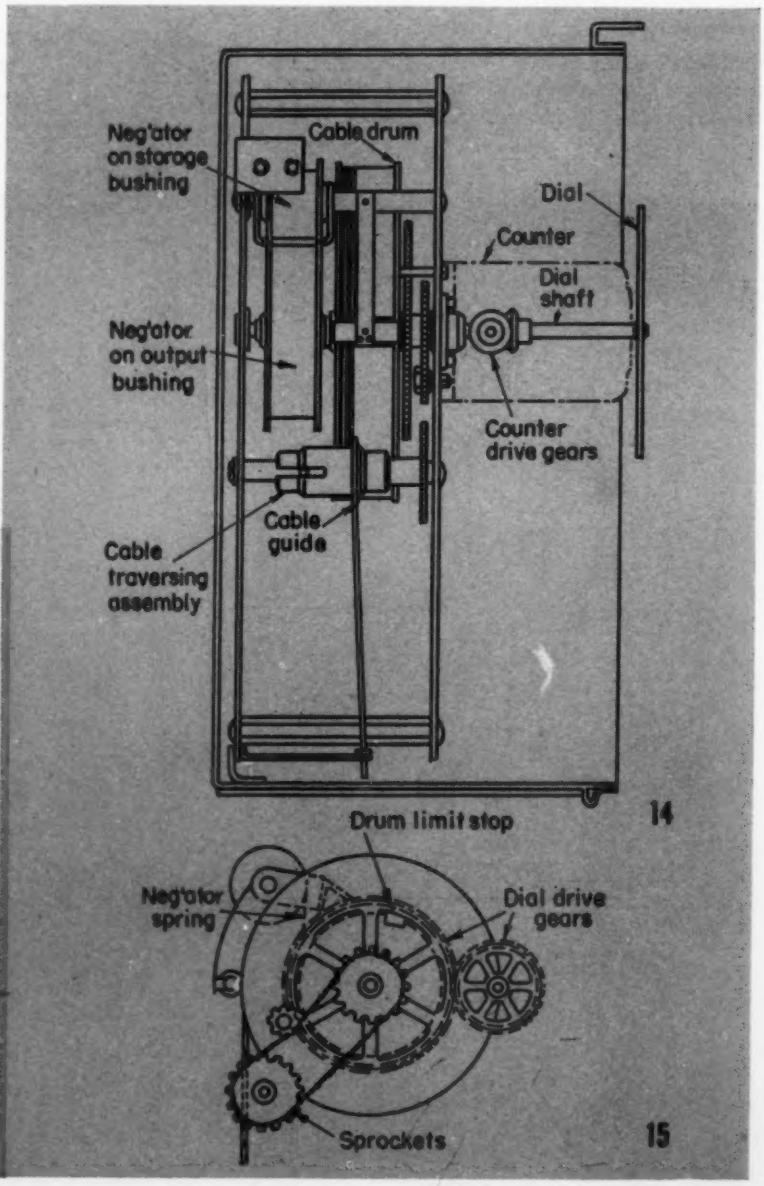

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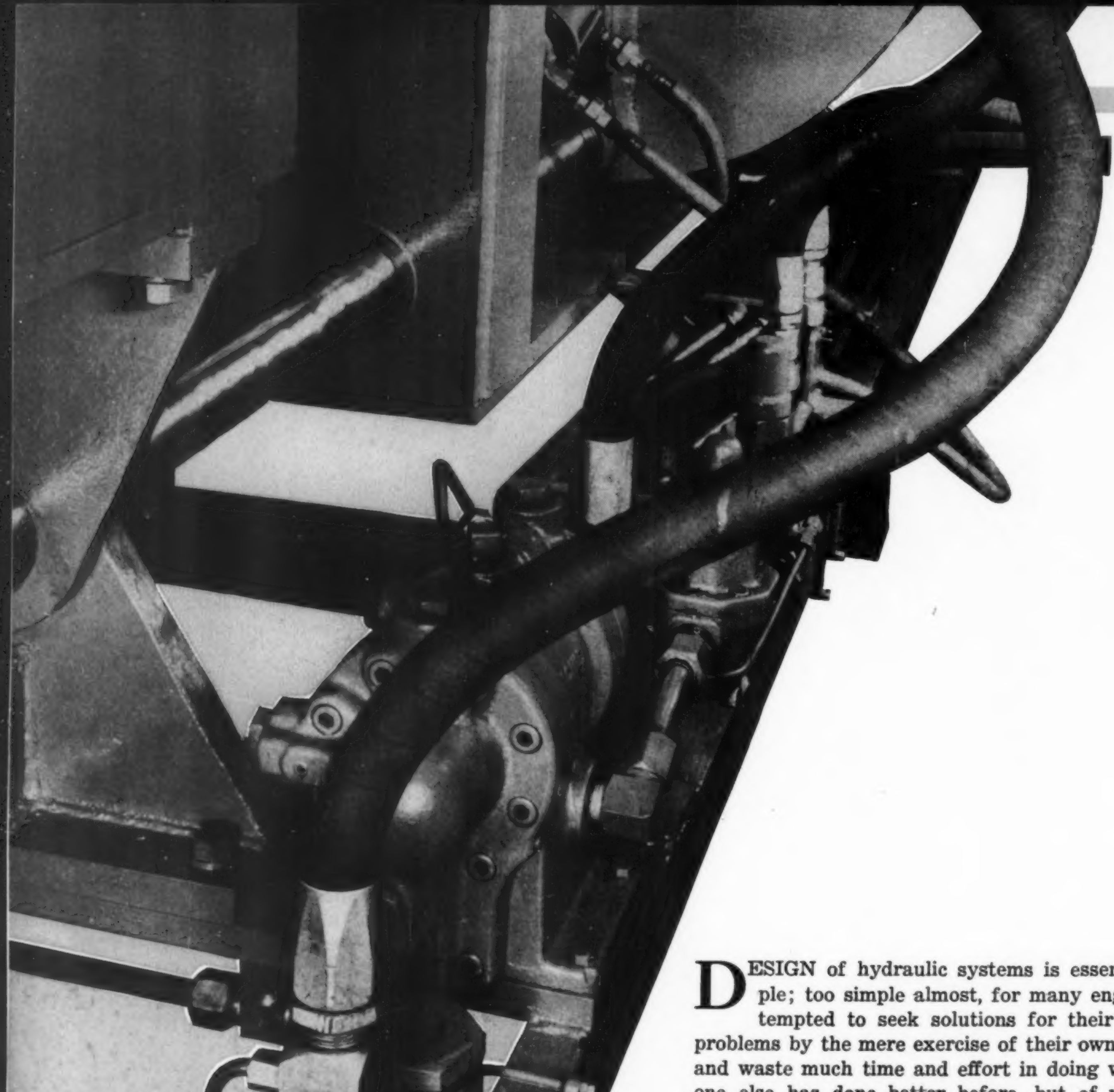
HYDRAULIC CONTROL SYSTEMS

By R. Hadekel
Consulting Engineer
London, England

Basic design of machine hydraulic systems is put on a methodical and consistent foundation in this series of articles to aid development of circuits for maximum simplicity and effectiveness



DESIGN of hydraulic systems is essentially simple; too simple almost, for many engineers are tempted to seek solutions for their particular problems by the mere exercise of their own ingenuity, and waste much time and effort in doing what someone else has done better before, but of which they were unaware.

Designing hydraulic systems and components which will work after a fashion is easy, but designing for maximum simplicity and effectiveness is not quite so simple. Besides experience, for which of course there is never a complete substitute, three things are necessary: (1) Acquaintance with the past practice in the art, (2) correct appreciation of the fundamental principles underlying all circuits, and (3) knowledge of the subtleties which often lurk behind the seeming simplicity of hydraulic equipment.

It is only quite recently that specific data[1, 2, 3, 4] have become available to help enlighten the practitioner on these matters. The endeavor here will be to put the subject on a methodical and systematic foundation. It is the hope that the reader who thoroughly assimilates this series of articles will approach his problems with full awareness of the possibilities and techniques of hydraulics, and will be able to solve them in an elegant manner and with a minimum of effort.

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[1]References are listed at end of article.

PART 1— GENERAL CIRCUIT PROBLEMS

Motion and Load: The concepts of *motion* and *load* underlie all machinery problems. Strictly speaking, the two concepts are inseparable, since no load can be generated without producing motion (be it only elastic deflection) and no motion can take place without load being exerted (be it only that required to overcome friction). In many cases, however, one of the two concepts becomes of minor significance. Thus in a plastic molding press the primary requirement is to maintain a given load on the dies while they remain practically stationary. In a remote signalling system the desired end is to reproduce motion at a distance, the load opposing this motion sometimes being negligible. In a lifting crane both concepts are equally important.

Motion may be transmitted, and load exerted, in many ways. In certain cases it is convenient to use a liquid as a medium for these purposes. Looking at the problem from the point of view of motion transmission, the convenience arises partly from the fact that the transmission line can follow arbitrary paths, and partly from the comparative ease with which motion can be controlled. From the point of view of load

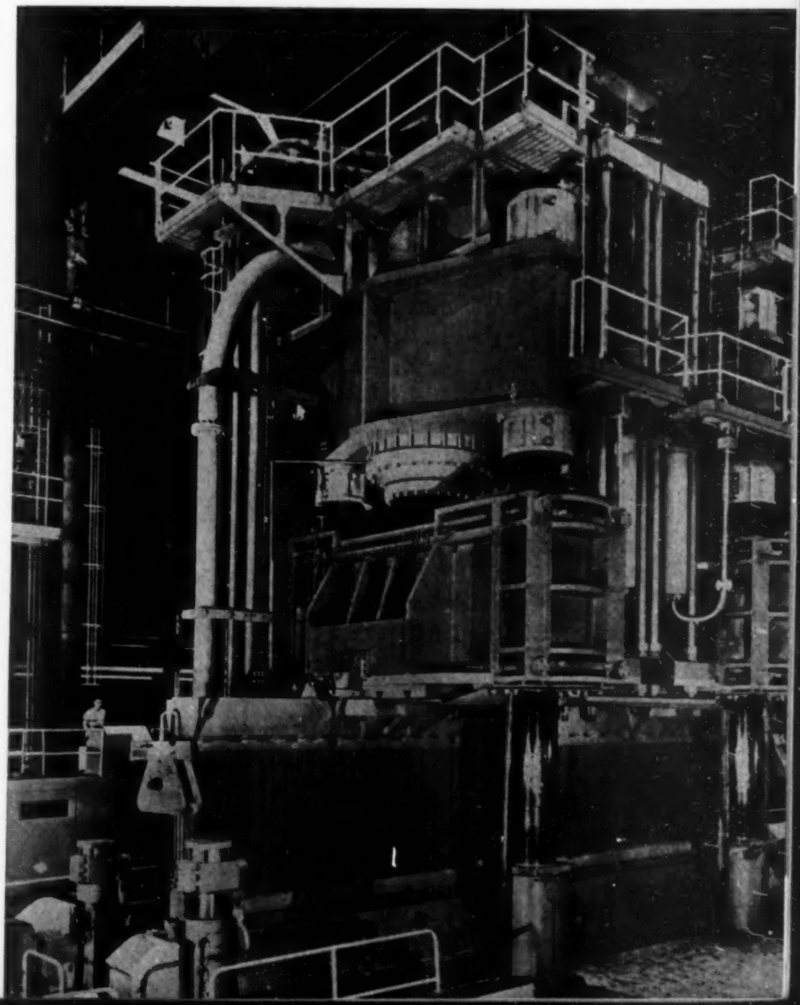


Fig. 1—Modern 6500-ton hydraulic press designed to hot pierce billets up to 26,000 pounds or cold form plate up to 2¾ inches thick by 10 feet wide and 42 feet long. Photo, courtesy Babcock & Wilcox Co.

exertion, the convenience resides mostly in the fact that loads of very high magnitude can be generated with ease. Among other factors may be mentioned ease of control of the magnitude of loads, ease of holding specified loads at standstill, and ease of proportioning loads in various members.

Applications in which loads are of such small magnitude that they could easily be exerted by a variety of different means form but a very small portion of the field of hydraulic machinery. The typical hydraulically operated machine is one which has to exert relatively high loads, i.e., in its "purest" form, the hydraulic press, *Fig.* 1. In the same category are all machines based on hydraulic cylinders which exert loads of some magnitude, usually at comparatively low speeds, e.g., aircraft undercarriage retracting gear, lifting cranes, etc.

In most applications the use of hydraulic operation in preference to other methods has been adopted for a variety of reasons, which often differ for individual cases. New applications are continually being devised and established ones are sometimes abandoned. For all these (and other) reasons, the scope of hydraulic machinery cannot be accurately defined, nor would any useful purpose be served by cataloging applications, especially since many significant examples will be found in the text as illustrations of general problems.

GENERAL DESIGN CONSIDERATIONS

Hydraulic System as a Whole: Guiding considerations in the design of circuits, and in the design or selection of equipment, are the same as those which apply to the majority of engineering problems. These can usually be classified as: (1) performance, (2) reliability, (3) safety, and (4) cost, although relative importance and even significance of all four are subject to wide variation and there is often no sharp dividing line between them.

The term *performance* has a particularly wide scope, and may comprise: efficiency in the strict sense of ratio of power output to input; operational effectiveness and convenience; compactness (often an important consideration), and, in the case of aircraft, low weight.

Reliability needs little comment, except perhaps to say that the term should include the ability to perform with the minimum of routine maintenance, and also easy rectification of troubles. *Safety* can be considered as freedom from danger to the operator (or occupant, etc.) during the normal functioning of the machine, which may be associated with freedom from danger to life and limb in the event of malfunctioning. In stationary machinery such danger rarely exists except from structural weakness, material or manufacturing defects, or overloading. In aircraft and in certain vehicle applications (e.g. steering and brake gear) reliability and safety are almost synonymous. Reliability should also include that of any methods of emergency operation which may be provided.

Cost must include that of installation, which may be an important item, and that of designing any special control gear or other apparatus which may be called for.

In the author's experience and opinion, the keyword to success in hydraulic machinery is *simplicity*. This comprises simplicity of the circuit, particularly in the sense of having as few valves as possible, simplicity in the functional principles involved, and mechanical simplicity of the individual components of the system. However, it is not considered justifiable to purchase simplicity at the cost of operational convenience. For instance, in a press which has to remain under load over an appreciable period of time, it seems wrong to call on the operator to ensure correct maintenance of the pressure when this can be done automatically, even if the system can be simplified by so doing. Likewise, in order to achieve simplicity it would not be deemed justifiable to call on the operator to hold on to a control handle if the natural and sufficient action corresponding to the functioning of the machine is to let it go. The author has seen these and similar inadequacies perpetrated, sometimes even with no apparent gain in other directions.

Commercial Considerations: The whole problem of

devising a suitable system for any particular machine is to some extent dominated by the fact that even standard commercially available hydraulic valves are expensive, coupled with the fact that design and manufacture of special valves for a particular machine would be more expensive still, unless quantities are very large. In order to avoid as much as possible being called upon to undertake the design and/or manufacture of special components, makers of hydraulic equipment endeavor to produce valves which are adaptable to a reasonable variety of applications, and which are therefore often not the optimum for a particular case. If, as is generally the aim, available standard equipment is to be used to the greatest practicable extent, it is often necessary to compromise, particularly on the score of simplicity.

Where production of a particular machine is planned in moderate quantities (say under a hundred, or even a few hundred) the use of available standard pumps, valves, etc. (but not necessarily cylinders, standardization of which is a difficult problem), is most often the only commercially practicable solution, ***Fig.*** **2.** On the other hand, where sufficiently large quantities are involved it is often economical as well as convenient to design special valves for the machine in question. Specially designed pumps are but very rarely justifiable.

Valve Design and Valve Groups: The more obvious aspects of the bearing of installation problems on valve design hardly need mentioning. They include suitable mounting arrangements (sometimes neglected), convenient disposition of connections from the point of view of pipe runs, and other commonsense considerations.

One point that is sometimes neglected is the desirability of avoiding external connections where these connections can be made within the valve itself. Thus if open-center control is required for a single cylinder, it is clumsy to use a six-way valve as in ***Fig.*** **3**, with two external connections at *A* and *B*, when a four-way valve could do the job. Similarly, a well-designed unloading valve for twin-pump systems should be as shown in ***Fig.*** **4**, the check valve together with connections *E* and *F* being within the valve body, external connections being at *A*, *B* and *C*, with preferably an additional connection at *D* to save a tee.

Certain other valves are almost invariably connected in a branch from a pipe line (usually the pressure main), the main types in question being relief valves and unloading valves. It is frequently the custom to design such valves with a double connection at the side corresponding to the branch, as shown in ***Fig.*** **5**, thus again saving a tee, with resultant reduction in cost, space, and possibility of leakage from joints. Multipurpose valves are helpful in making neat installations and reducing cost, ***Fig.*** **6**.

For installation on mass-produced machines, it is often convenient to group a number of valves (and possibly also other items such as small accumulators) together and mount them on a separate special bracket, ***Fig.*** **7**, the necessary interconnections within the group being made before the bracket is assembled on to the machine so that a minimum number of connections need be made on final installation. Such a procedure is often further improved by the use of "pad mounting" valves in which all connections (with possible exceptions, particularly for pilot lines) are made on one face, the connection holes being unthreaded and formed with a recess to receive a rubber ring designed to seal against a flat mounting pad, ***Fig.*** **8**. One such pad or transfer plate receives a number of valves, connections being established, before the valves are mounted, by pipes at the back of the pad or by drilled

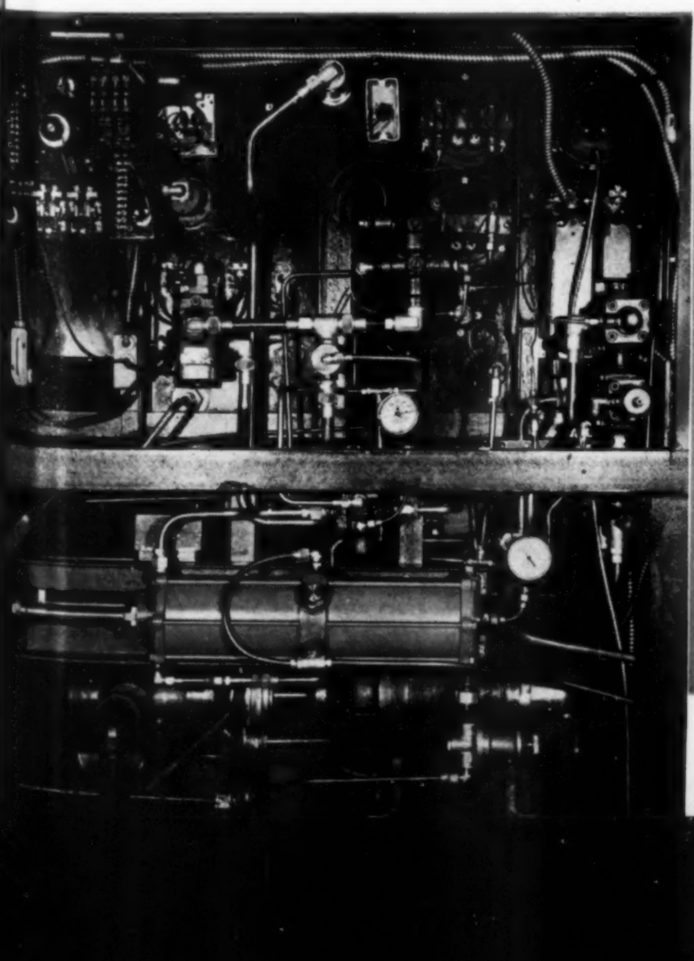
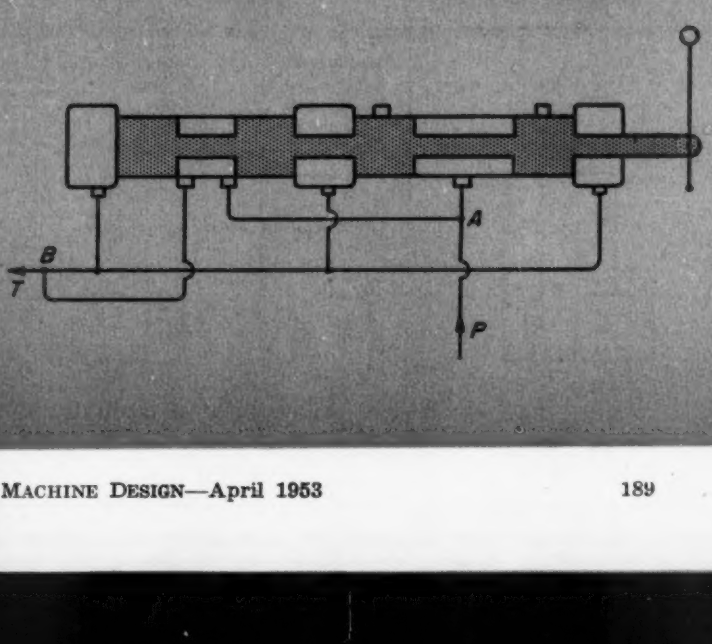


Fig. 2—Left—Use of standard available components in hydraulic circuits is generally the best commercially practicable approach

Fig. 3—Below—Special six-way valve is less desirable than a standard four-way available to accomplish a similar function



SYMBOLS FOR HYDRAULIC C

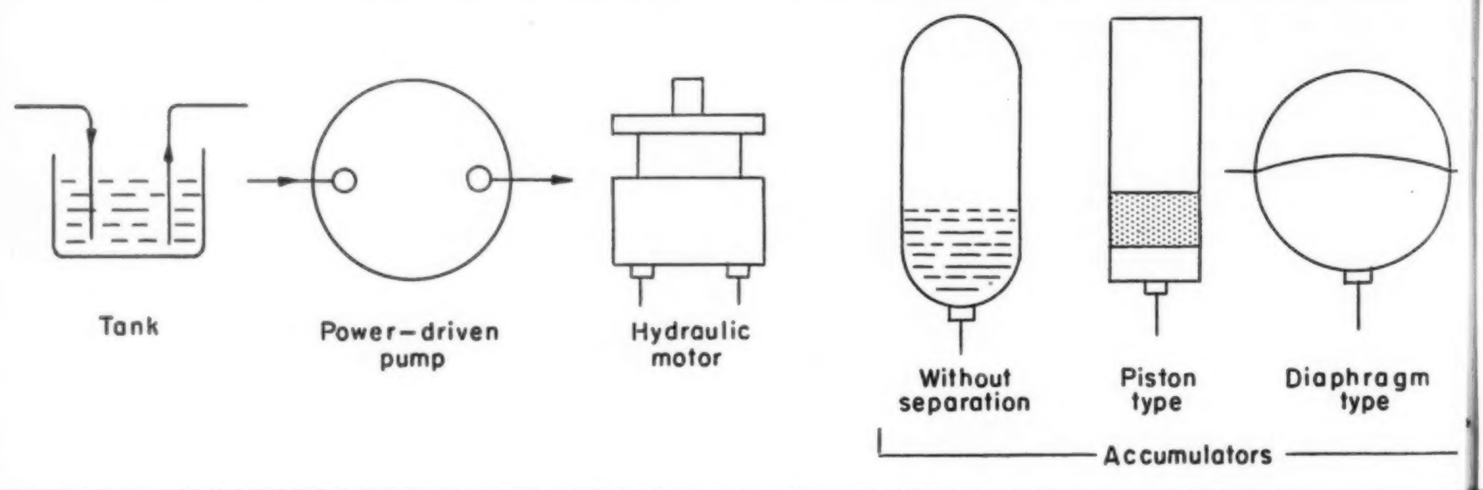
Until comparatively recently, every organization had its own ideas on hydraulic symbols. Because many practices were prevalent, diagrams were difficult to follow. For example, two parallel lines represented a pipe, when a single line would do and look much neater. Drawings of components were overelaborated, showing minor constructional details. Purely formal representations of components were made in the form of their external outline or just by a rectangle, with no clue as to their functions.

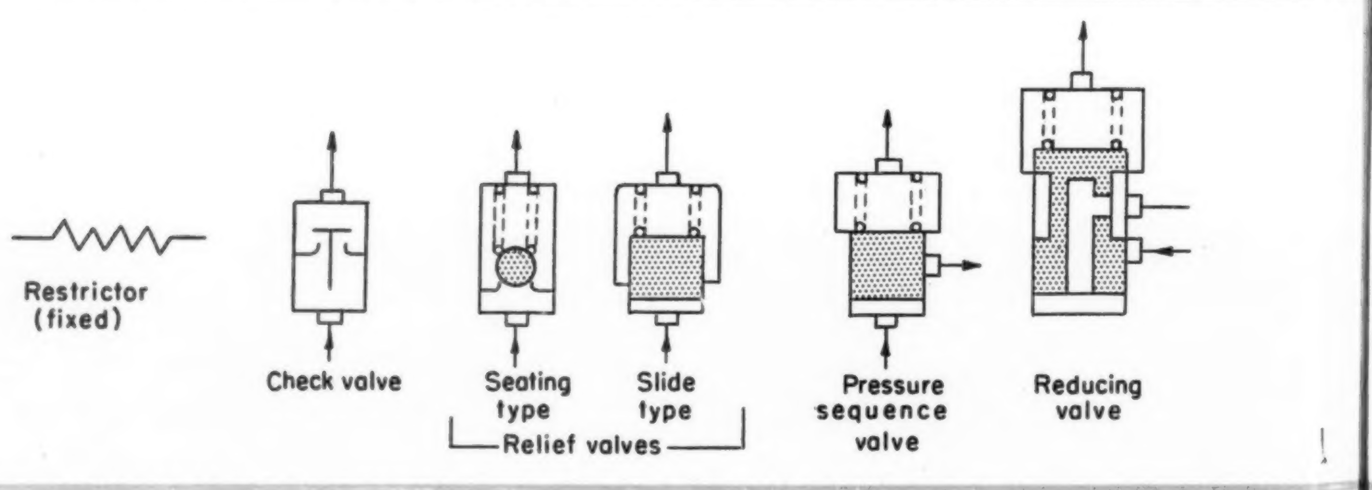
In the United States the subject has been settled largely by the adoption of the JIC system of symbols. In Great Britain a standard system of symbols was adopted by the aircraft industry during World War II and this system has been used for the diagrams illustrating the present series of articles. Unfortunately this system has not as yet been widely publicized but its general principles are as outlined in the following paragraphs.

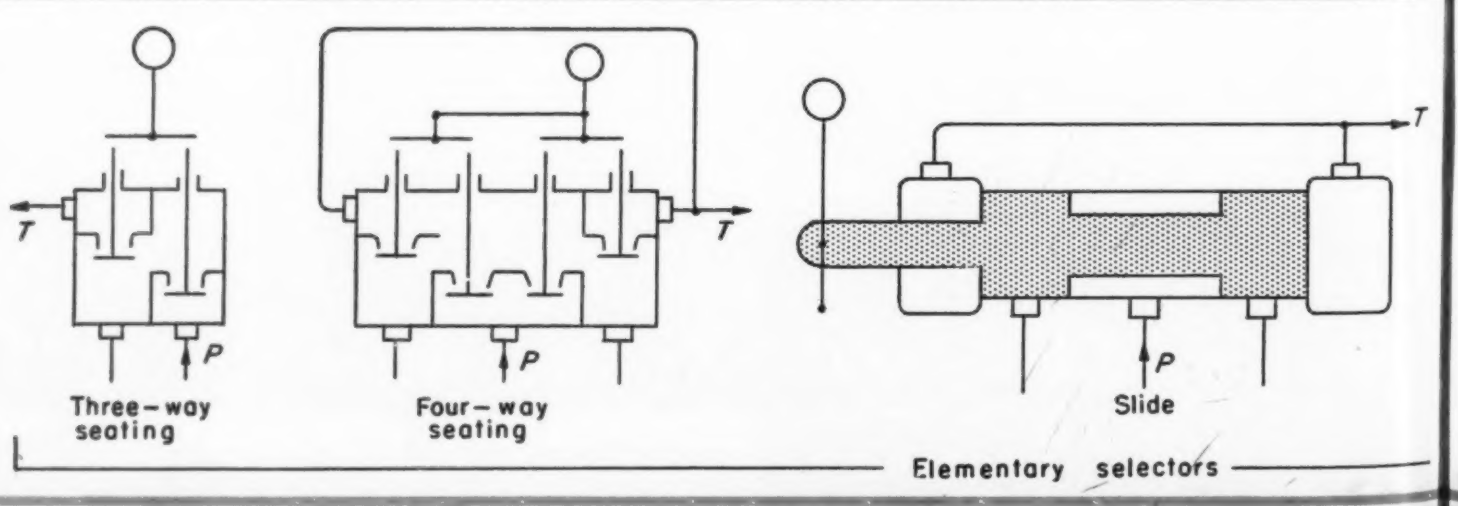
1. Pipes are represented by single lines, which may carry an arrow to indicate the direction of flow, if this is always in the same sense. In addition, the letter *P* indicates a line from the pump, *T* a line to the tank.

2. Symbols for relatively complicated components such as pumps are purely conventional.

3. Symbols for valves and other simple components are such that their function and mode of operation are plainly visible. In simple diagrams the function only is considered and, say, a rotary valve symbol may be used for the sake of simpli-





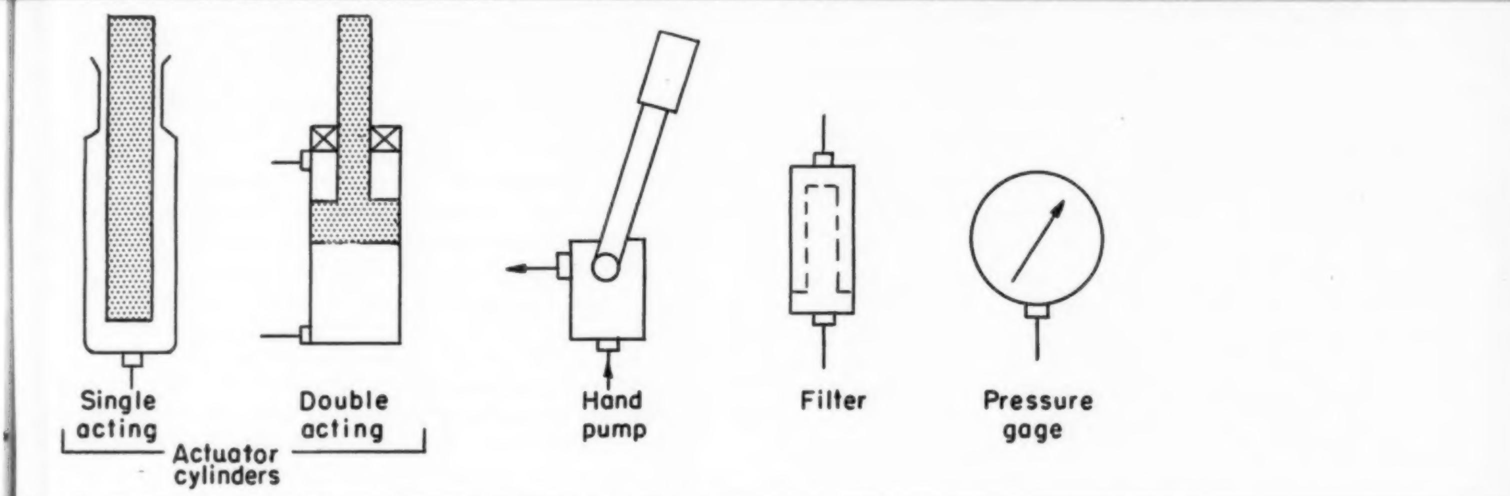


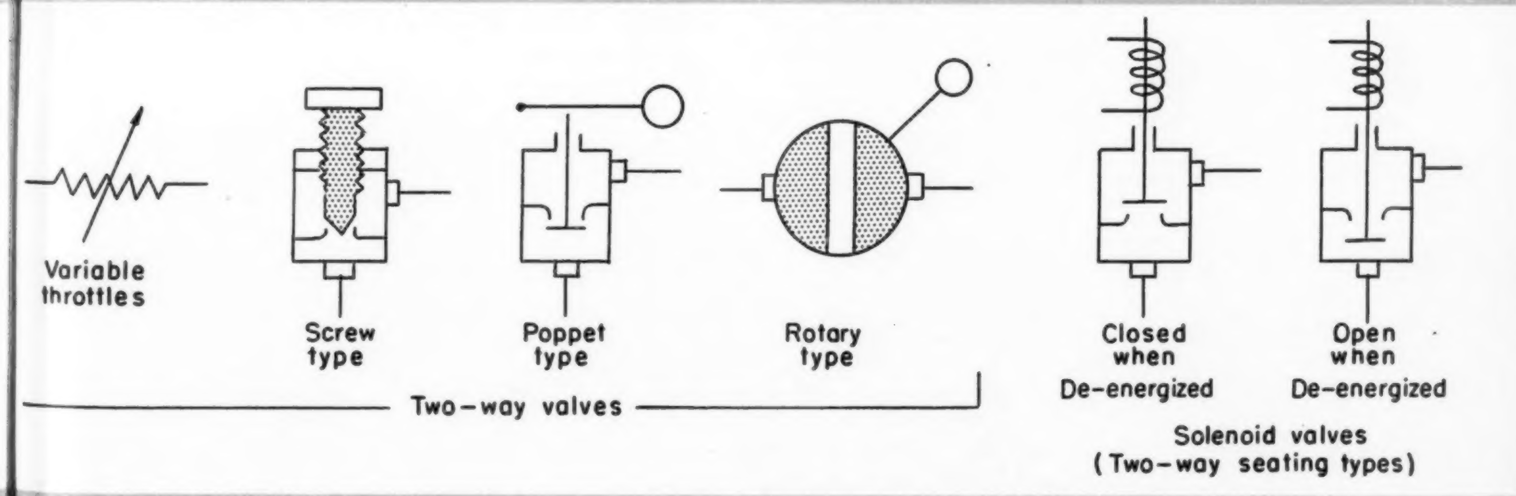
city. In more detailed diagrams of actual systems the symbol will be more explicit, and will show whether the valve is of the rotary, slide or seating type and its actual basic arrangement to the exclusion of constructional details.

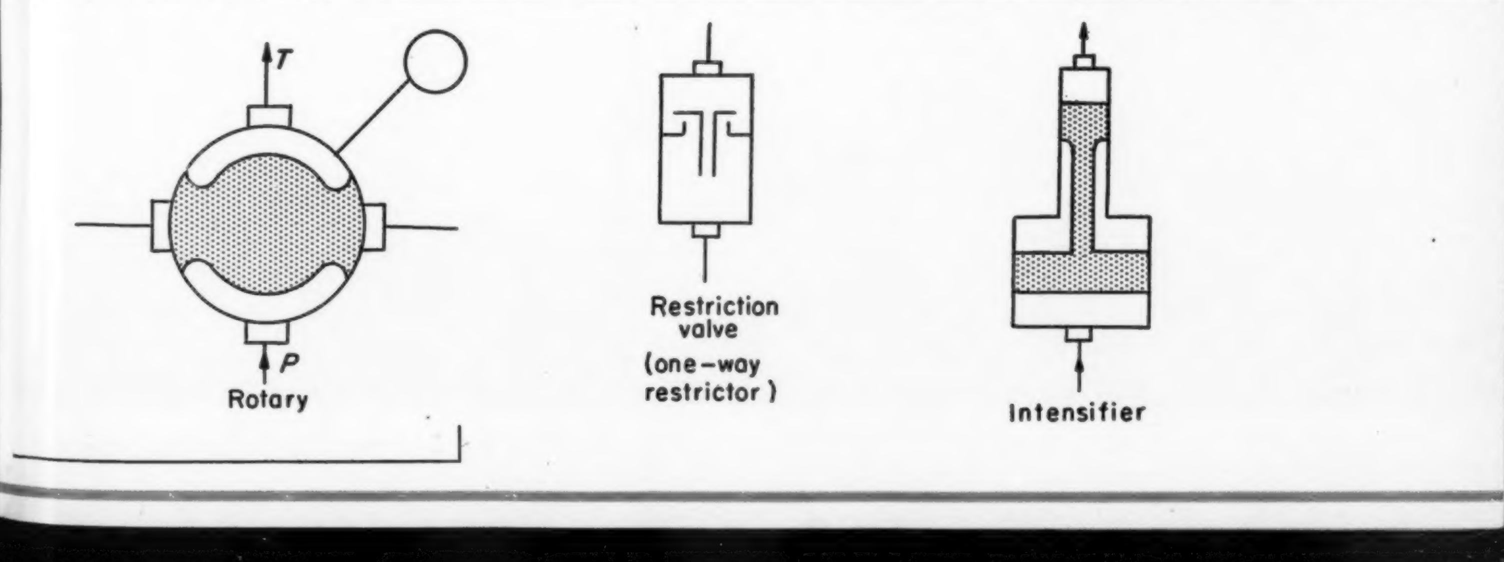
4. No attempt is made to evolve a fully comprehensive set of symbols for valves, since their variety is practically limitless and new types are continually being devised. Adaquate symbols for new types are easily developed by following the same principles as those used in existing symbols, as shown on this page.

Certain additional recommendations are worth mentioning, although not strictly relevant to the present text. These are that there should be three kinds of diagram: (1) A simplified one illustrating only the mode of operation, and omitting details such as filters, pressure gauges, etc., and showing only one of a number of cylinders in parallel; (2) A complete diagram showing every component and pipe, the items being roughly laid out according to their positions in the aircraft, vehicle or machine; and (3) a perspective drawing illustrating the positions of the components when installed. The complete diagram also indicates pipe sizes, output of pumps, pressure setting of relief valves, etc.

The accompanying diagrams show the symbols for certain of the more important valves and for components other than valves. Symbols for other valves are introduced when the particular type is discussed in the text.







passages within it, *Fig.* 9.

Valve grouping and reduction of piping may be carried to the extreme in those cases where the control gear is specially designed for one particular application. In such cases a number of valves may be located within a single body, and all their interconnections achieved by internal drillings within that body.

Valve Size and Capacity: Valve size is usually specified by the size of the connections. There is of course no absolute reason why the several connections should be the same, but for standard valves it is the rule to make them so, with the exception of pilot connections. Another exception is often made in the case of selectors intended to control single acting (gravity return) presses where the pressure available for return is usually extremely low, necessitating large passages and pipes in the paths from the selector to the cylinder and tank although a much smaller pipe is sufficient for the pressure inlet.

Passages within valves are usually made of approximately the same size as the pipes for which the connections are intended. Definite flow rating, i.e., a maximum permissible flow figure, usually has little significance for a valve unless the maximum system pressure is specified, since most valves cannot be "overloaded". The usual criterion of permissible flow is the permissible pressure loss through the valve, which is generally set at a given fraction of the maximum system pressure. One exception arises in the case of valves which are sensitive to back pressure in the return line, as with many hand-operated slide valves return line pressure would tend to move the spool. Another exception arises in certain automatic valves, particularly relief valves, which may have a definite maximum flow capacity.

Standard Valve Ranges: There are comparatively few manufacturers who produce really extensive ranges of valves for industrial hydraulic machinery. A desirable range of valves should fulfill the following requirements:

1. Valve types should be available in a number of sizes to cover the requirements of machinery of varying horsepowers, without making it necessary to use valves much larger than the minimum acceptable. For industrial purposes (not aircraft), it should be sufficient to standardize say on ¼, ½, 1, 1½, and 2-inch sizes, omitting intermediate steps. The call for sizes larger than 2-inch is extremely rare. The ¼-inch size would be used in pilot lines, and also in high-pressure systems (say, over 3000 psi) of comparatively low power.
2. Within the limitations of commercial practicability, the range should come as near as possible to including every type of valve that is likely to be called for in the solution of any circuit problem which may recur with reasonable frequency.

Hydraulic equipment calls for comparatively elaborate and expensive tooling, which can be justified only if production is large enough and this factor usually sets a limit on the variety of standard valves which a manufacturer is prepared to produce. Ingenuity in design can be effective in connection with this problem, as it is possible to produce a wide variety of valve types from a comparatively small number of detail components used in different combinations, provided the range is designed as a whole from the start. Thus, many different types of selectors can be made up from comparatively few basic parts; similarly a range of automatic valves can have a large proportion of components in common.

Determination of Pipe and Valve Sizes: For any given machine, loads and speeds of operation being specified, the rate of flow is determined when a value

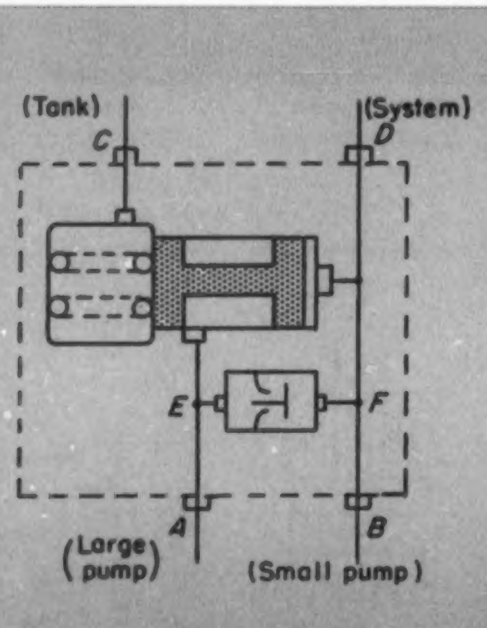


Fig. 4—Left—Unloading valve with internal check designed for a twin-pump circuit

Fig. 5—Below—Typical design for relief or check valve

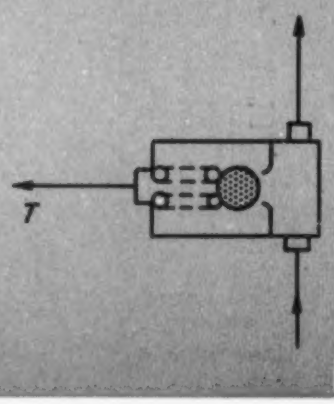
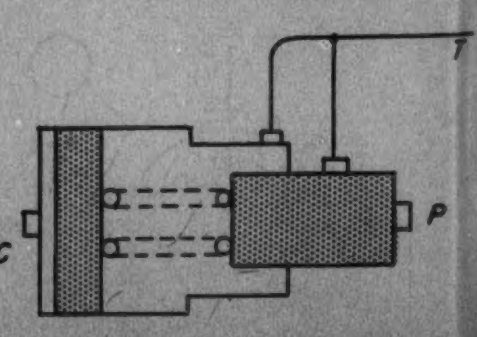


Fig. 6—Combination unloading and relief valve designed for circuit simplification



is assigned to the maximum pressure. The latter can usually be chosen arbitrarily within limits, and the factors governing its choice will be discussed later. It will be assumed here that some suitable value has been chosen and, hence, that the flow is fixed.

A few valves have a definite flow rating, and with these the minimum size is immediately determinable. In general, however, the criterion for both pipe and valve size is pressure loss.

Rule-of-thumb criteria for permissible flow (based on a fixed value of allowable linear velocity) are to be strongly discouraged as being of little significance. No doubt the soundest method is to fix an arbitrary allowable pressure loss, and calculate minimum pipe and valve sizes accordingly. The significant quantity from the efficiency point of view (which is the usual criterion) is of course not the absolute loss, but the loss expressed as a percentage of maximum pressure. The choice of a figure for this value may vary with different applications, according to the relative importance of power economy and installation compactness. For most industrial applications, a figure of 5 per cent of the maximum pressure seems reasonable.

There are two sources of loss, viz., pipe friction, and orifice effects. The latter include not only direct effects of restriction (which may often be negligible), but also the effects of *sharp* changes of direction of flow. There are usually several such changes of direction within a valve and similar changes occur at tee pieces and "banjo" connections. Hence, passages in valves should be of the same order of size as the maximum bore of the pipes which the valve is designed to take to permit a single orifice loss criterion to cover both pipes and valves (in the smallest sizes, orifice effects are usually low as compared with pipe friction effects and valve passages may be made rather smaller than the bore of the pipes).

In practice one of the two factors (orifice loss and friction loss) is almost invariably considerably higher than the other and it is therefore sufficient to determine which of the two is critical, and calculate sizes on the basis of that effect only, neglecting losses due to the other.

Assuming that the sum of all the orifice effects in a system is equivalent to the discharge loss through n orifices of diameter d in inches equal to the pipe bore. If C is the orifice coefficient and p the permissible total loss in psi, the allowable flow q in gpm is given by:

$$q_3 = 30.5\, d^2\, C \sqrt{\frac{p}{n}} \qquad (1)$$

The pipe friction loss will depend on whether flow is laminar or turbulent, the transition between the two being approximately at the point where

$$\nu = \frac{q}{76\, d} \qquad (2)$$

ν being the kinematic viscosity in stokes.

For laminar flow the maximum viscosity ν employed is that corresponding to the lowest temperature for which the given loss can be tolerated. The permissible flow is then given by

$$q_2 = \frac{36\, p\, d^4}{L \nu_{max}} \qquad (3)$$

where L is the length of pipe run in feet.

Under turbulent flow conditions two cases must be

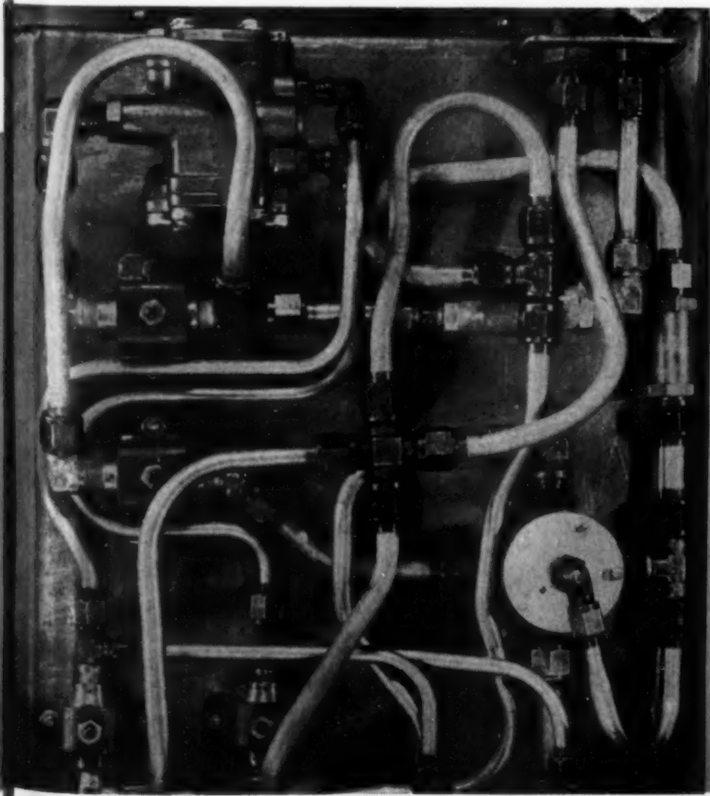
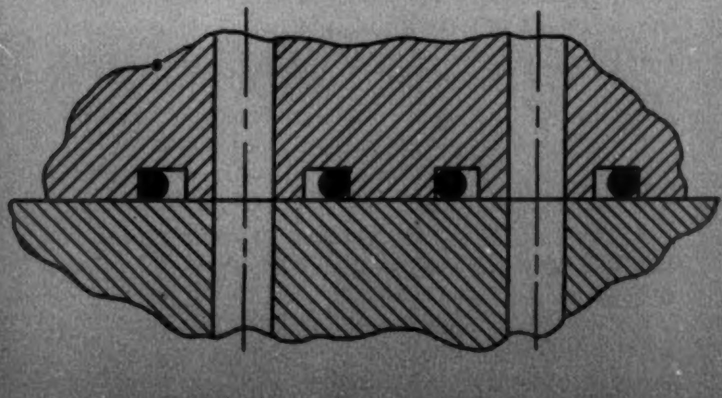


Fig. 7—Left—Groups of components can be preassembled on a special panel for ease in final assembly

Fig. 8—Below—Typical method of sealing ports of pad-mounted valves



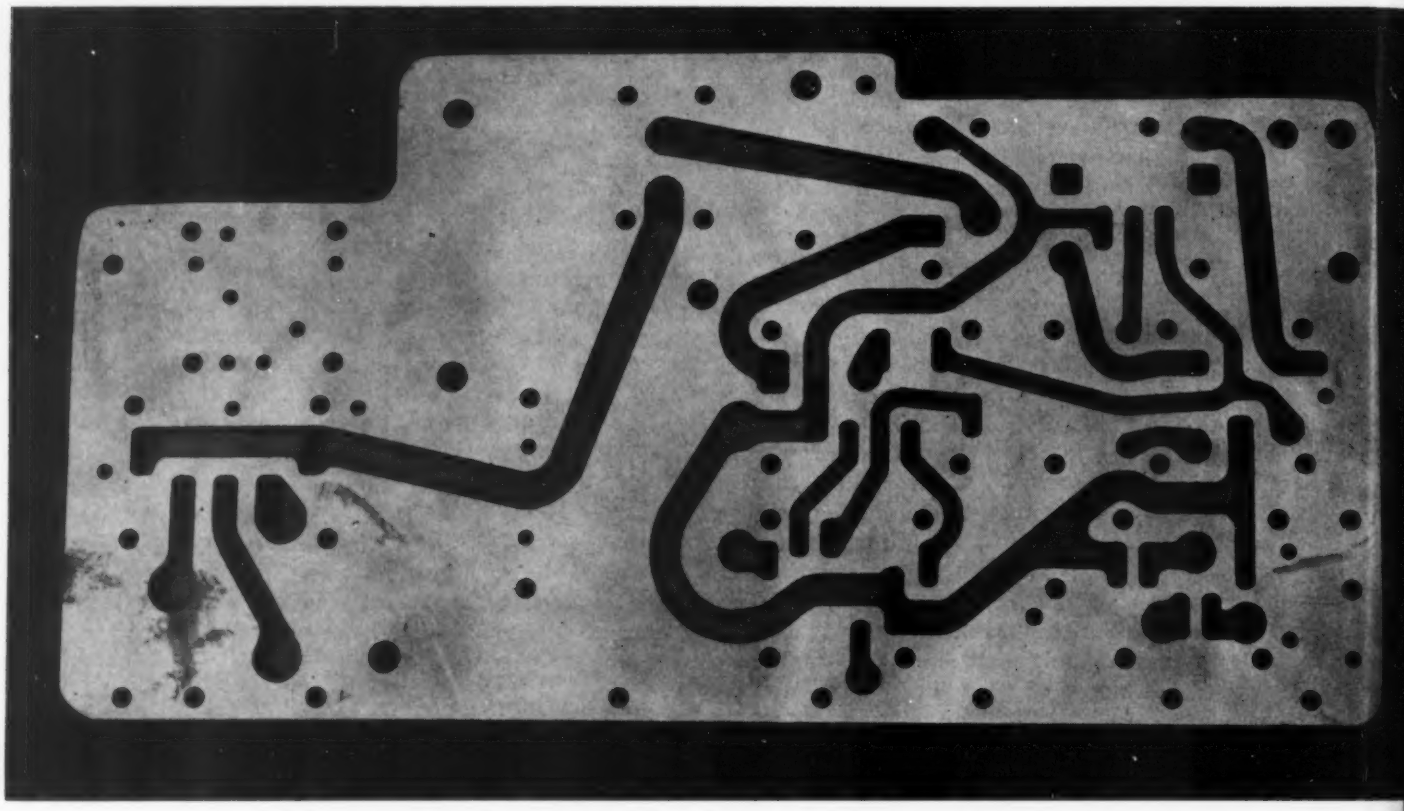


Fig. 9—Transfer plate for a Heald internal grinding machine. Cored passages provide flow channels for pad-mounted valves to eliminate piping

considered. If either laminar or turbulent flow can occur according to the temperature, i.e., if the transition point lies within the working viscosity range, then the maximum loss will be at the transition point temperature and viscosity. The permissible flow is then given by

$$q_1 = 40.4 \sqrt{\frac{p\,d^5}{L}} \quad \text{(4)}$$

If flow is turbulent for the full range of temperatures under consideration, the highest value of viscosity must be taken and the permissible flow is given by

$$q_1' = \left[\frac{550\,p\,d^{4.75}}{(\nu_{max})^{0.25}\,L}\right]^{4/7} \quad \text{(5)}$$

From Equations 1 through 5 it is possible to derive the following criteria:

1. Orifice losses are never critical if the flow is less than

$$q' = \frac{25.8\,C^2\,\nu\,L}{n} \quad \text{(6)}$$

2. Turbulent flow conditions are never critical if permissible pressure loss is greater than

$$p' = \frac{6.78\,\nu_{max}^2\,n^3}{C^6\,L^2} \quad \text{(7)}$$

3. If turbulent flow conditions are critical, they can only be so between values of pipe length given by

$$L = \frac{1.18\,d^2\sqrt{np}}{\nu_{max}\,C} \text{ and } L = \frac{nd}{0.571\,C^2} \quad \text{(8)}$$

The temperature range will depend on the application. For industrial machinery working indoors, a range of 20 C to 60 C seems reasonable, lower temperatures being disregarded on the grounds that comparatively high losses can usually be tolerated while the system warms up (except for suction pipes). For suction pipes somewhat lower temperatures should be taken, say 10 C for this type of application considered. A loss of 2 psi is suggested as reasonable. Orifice effects should be negligible with a well-designed installation.

This theory is applicable to systems in which the flow is derived directly from the pump. Where use is made of a large capacity accumulator recharged during idle periods, somewhat different criteria may be taken, especially if peak loads occur only during a comparatively short fraction of the actual working time. With an accumulator, it is the pipe size that determines the speed of operation and sizes must be calculated accordingly. Pipes can often be substantially smaller than with systems operating directly from a pump.

Choice of Working Pressure: The chief factor favoring high working pressures is reduction in size of cylinders, valves, pipes, and other components and also the consequent reduction in cost, except for the pump.

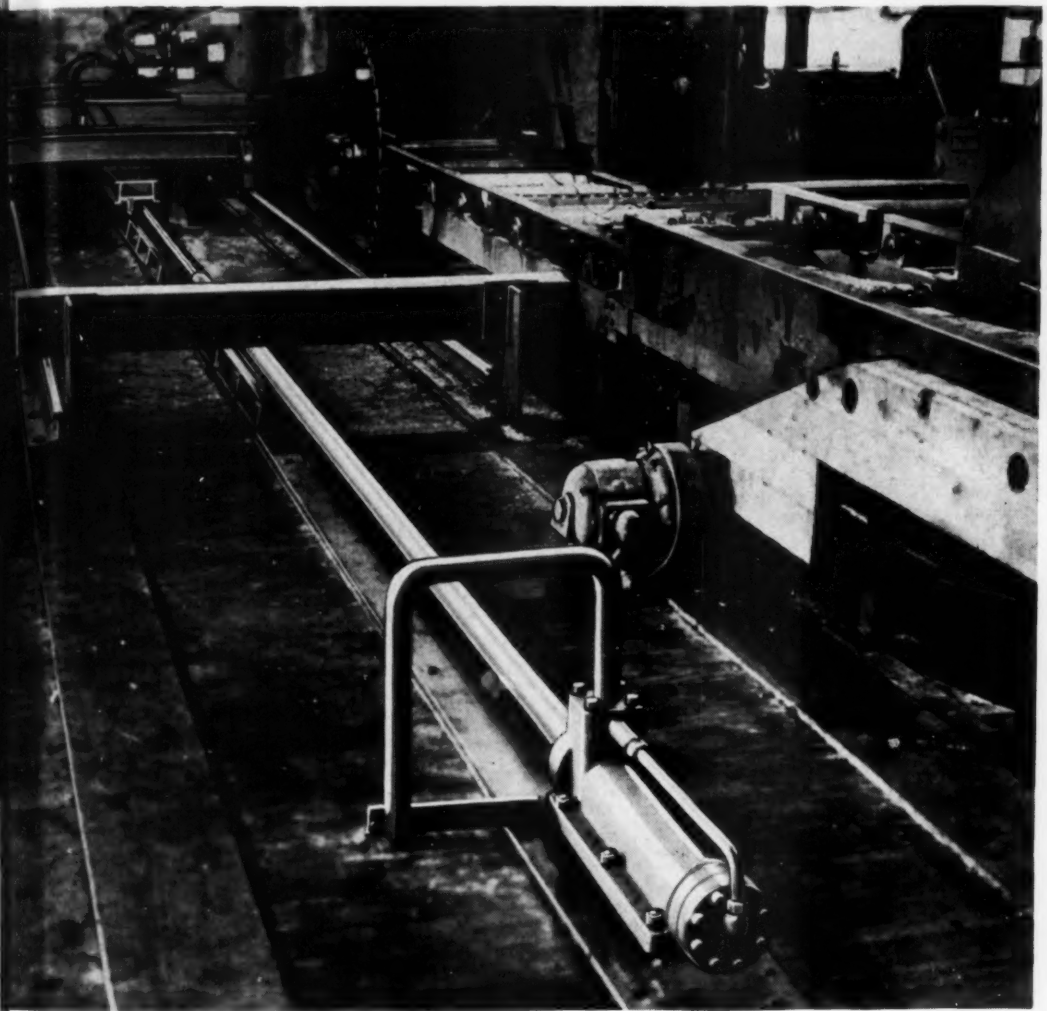


Fig. 10—Long-travel hydraulic cylinder used for a sawmill table drive carries maximum work load under tension, returns unloaded in compression. Photo, courtesy Oilgear Co.

Pump cost is generally a function of horsepower rather than delivery. Elasticity effects are also of importance, in some cases, and their magnitude increases with increasing pressure. Consider for instance a cylinder designed to exert a maximum load F in pounds over a stroke s in inches. If the maximum pressure is p, in psi, the cylinder area is F/p and volume Fs/p. If K is the bulk modulus of elasticity, the volumetric compression of the oil in the cylinder is $Fsp/pK = Fs/K$ and the energy stored in the compressed oil is $Fsp/2K$, i.e., directly proportional to the magnitude of the working pressure selected.

Elasticity effects may be unfavorable in that they increase the power required to build up to full load in a given time, the criterion of power in certain presses. Too, they may be favorable if they are to be utilized as the source of capacity in an unloaded live system, dispensing with an accumulator.

A limitation is often set by the strength of a cylinder, if the latter is of substantial length, and high loads occur in compression, *Fig.* 10. Thus, for a single-acting cylinder with an effective area equal to that of the piston rod, a minimum size and hence a maximum pressure is immediately obtainable from the buckling load. With double-acting cylinders the piston rod size may be determined similarly from the buckling strength, which may limit severely the gains to be obtained by increasing pressure beyond a certain value.

In the case of aircraft systems, weight is the major consideration although size is almost equally important. On the score of weight, high pressures appear to be beneficial, although there is probably an upper limit beyond which weight would increase again with increasing pressure.

High pressures are generally favorable from the point of view of operating loads of valves, provided of course that the minimum size of valve is chosen according to the pressure. Speed control is easier if pressures are comparatively low (especially if the power is low), as it is difficult to exert accurate control if throttling requires passages of very small size. This consideration applies particularly to applications in which speed is controlled according to selector position, e.g., handling gear and aircraft gun turrets.

In large presses the overriding consideration is usually the size of the ram, and hence the highest practicable pressures are often used, say, 6000 psi or even more. In general, the benefits to be derived from high pressures increase with the horsepower.

On low-power applications in certain machine tools, handling equipment, printing presses, etc., there is often little to be gained by going beyond 1000 psi,

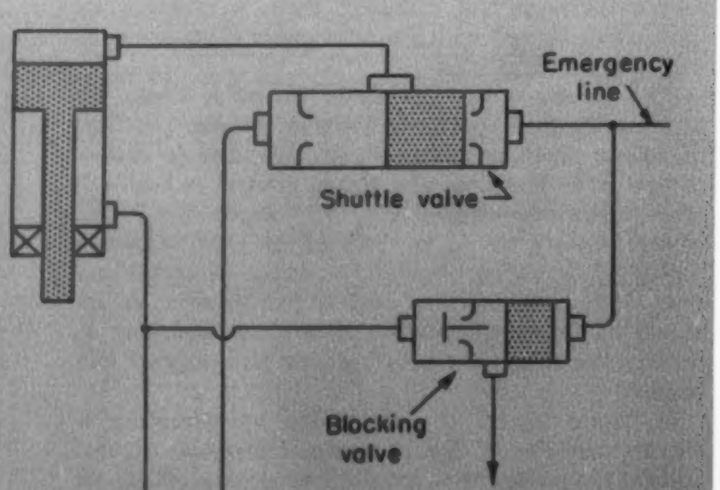
and even much lower pressures are sometimes used. For small presses, broaching machines, fork lift trucks, and other medium power applications in present-day practice, pressures vary from 1000 to 3000 psi, usually rising with increasing power.

An important consideration is the availability and cost of suitable equipment, particularly pumps. For a given horsepower the price of pumps has some tendency to increase with increasing pressure. For low or medium pressures it is possible to use simple and consequently low-cost rotary gear or vane pumps which up to now have been made to work at pressures up to 2000 psi or slightly higher. Piston pumps, which are the rule at present for pressures above 2000 or 3000 psi, tend to be rather expensive. Though comparatively simple, gear and vane pumps are a high-precision product if designed to give a good performance at pressures higher than about 500 psi, and it is only for pressures below this approximate figure that really low-cost pumps are now available.

Stress Problems: The safety factor problem assumes a special aspect in hydraulic equipment, since with certain reservations loads are limited to an accurately known figure by the relief valve or other pressure-limiting device. For this reason comparatively low safety factors can generally be used. The main reservations are:

1. With locked columns of fluid, pressures above relief valve setting may arise from external loads or, unless safeguarded, from thermal expansion.
2. Inertia effects may cause high pressures if a cylinder is stopped at midtravel.

Fig. 11—Circuit employing a shuttle valve to permit emergency pressure, through a blocking valve, to release return side fluid from the cylinder



The second effect is of importance only in comparatively few cases, since in most hydraulic machinery either stopping in midtravel is never called for, or the masses moved are comparatively small in relation to cylinder loads. Where inertia effect does arise, the pressures generated can be effectively limited by suitable cushioning devices.

Current British requirements for aircraft hydraulic equipment stipulate general factors of 2 and 1.5 (based on ultimate strength), referring respectively to the maximum pump pressure and the maximum pressure generated in locked columns by external loads or thermal expansion. For pipes and pipe couplings the factors are raised to 3.0 and 2.225, respectively. For proof tests both on pipes and other components, the factors are 1.5 and 1.12, respectively.

For applications other than aircraft there seem to be no generally accepted standards. In the author's opinion a general factor of 3, and a factor of 4 on pipes, should be adequate, provided due consideration is given to possibilities of mishandling. Formulas for hoop and longitudinal stresses in cylindrical members, and stresses in flat plates subject to pressure, are too well known to need repeating here.

In general, design of hydraulic equipment involves few specialized stress problems; perhaps the only one worth discussing is the column strength of cylinders. A cylinder can be considered as a strut of varying section having the dimensions of the piston rod over part of its length and of the cylinder for the remainder. Various methods of calculation have been proposed but the author recommends the following, which has the merit of extreme simplicity and has been found to work well in practice besides being based on theoretical considerations. The method applies to a cylinder pinned at both ends:

1. If the moment of inertia of the piston rod section is more than one quarter of that of the cylinder section, and/or if the protruding length of the rod is substantially more than half the total length, no serious error will be involved in treating the unit as a strut with a section equal to that of the piston rod and with an effective length equal to the total length. Any error will of course be on the conservative side.
2. In other cases, it can be shown that the cylinder is equivalent, within quite close limits, to a strut of section equal to that of the piston rod, but having an effective length of 80 per cent of the actual length between pin centers.

In either case, the usual column formulas are applicable.

General Principles of Component Design: In the majority of hydraulic components, except continuous pumps and motors, the significant motions involved are rectilinear translations and major loads act along the central axis of the moving member. Such conditions are ideal from the point of view of mechanical design, since parasitic transverse loads are eliminated together with their attendant friction and wear problems.

It is sound design practice to aim at obtaining such conditions wherever possible, or at least to approximate them. Levers, cams, etc., should be avoided as long as there is an equally good solution without them. More particularly should transverse loads on sealed members be eliminated, if possible, either by providing guides outside the seal housing or (as in the case of cylinders) by mounting between pins. Better still is mounting between universal joints in order to avoid misalignment effects, even if motion is confined to one plane.

Misalignment and eccentricity errors are frequent causes of parasitic loads and other troubles particularly to be guarded against in sealed members because of the comparatively low clearances involved. Where a member is guided in two distinct and nominally concentric surfaces, it is sometimes possible to avoid eccentricity and misalignment effects by separating it into two portions, each of which can align itself on its own guide. If both portions have seals relying on close fits only, any other solution is to be strongly discouraged. Otherwise, detail design should be such that concentric diameters can be machined at one setup. Centering on screw threads is not good practice, and can be avoided by providing a plain shoulder register for location. The thread should have sufficient clearance to allow the register to be effective; better still, the two components to be connected can be held together by a separate screw or nut.

Provision should be made to avoid the danger of cutting soft seals during assembly, which arises when the seal must enter into a bore having a sharp entry, or when the seal has to pass an opening in the side of the bore. In the first case, the bore should have a chamfer at its entry. A radius is still better but is somewhat more expensive. In the second case, the bore should be enlarged at the opening and the recess should be designed with chamfers at both ends. It may also be possible to radius the entry of the opening in the bore.

Pump Installation and Suction Problems: In theory, the position of a pump in relation to the tank is of no importance as long as (1) the suction line is not sufficiently long to cause excessive pressure losses, and (2) the height of the pump above the tank is not great enough to cause serious loss. Thus, in static machinery it is often convenient to mount the pump and its driving motor on top of the tank. With the pump above tank level, if there is an imperfect joint in the suction line, resulting in the admission of air, depriming will occur when the pump is not running. With careful piping, this danger is seldom worth worrying about.

Again, if the pump is above tank level depriming *may* occur (still under stationary conditions) if there is admission of air elsewhere than in the suction line, but only if the difference in head between pump and tank is sufficient to break the capillary seals within the pump. Air may come in (1) through an imperfect joint in any part of the system, or (2) through the leakage return line (if the pump has one) if the line ends above the oil level in the tank. There is of course no reason why the line should not be set lower.

In a few installations it may be found impracticable to locate the pump near the tank and to provide suction lines of sufficient size to compensate for their length. In such cases some form of suction boost may be provided. One way of doing this is to pressurize the tank—a solution which has little to commend it, although simplifications have been proposed in which the boost pressure is automatically created by an air injector device operated from the return flow. Another solution is to create the boost by a second (low pressure) pump located near the tank. This pump may be electrically driven and may be of the centrifugal or positive-displacement type. In the latter case the delivery must be slightly higher than that of the main pump, the excess being blown off through a low-pressure relief valve.

Solutions have also been proposed in which the booster pump is driven by a hydraulic motor connected in series in the main pump delivery line. This scheme has the merit of providing automatic control for variations (if any) in the speed of the main pump, but it is practicable only if the main delivery line must pass near the tank in any event.

In some installations the pump is mounted within the tank and submerged in the oil. This is a neat and efficient solution, though not ideal from the point of view of servicing or seal replacement.

In most stationary installations pumps are driven by electric motors, and the power rating of the latter needs consideration. The rated power of electric motors is usually much less than their peak power, and it may be wasteful to choose a motor of rated power equal to the peak power of the pump, if the latter is required only for short periods. Ernst[2] recommends basing the rated power of the motor on the root-mean-square power absorbed by the system, as calculated from the working cycle, with an overriding provision that the motor stalling torque must of course exceed the peak pump torque.

Emergency Provisions: Provision for retaining the ability to operate certain cylinders (usually in one direction only) in the event of failure of certain components or lines is largely confined to aircraft systems. These generally include a hand pump to provide against failure of the main power-driven pump, and usually provide a safeguard against failure of certain lines or even valves.

Failure of lines is usually guarded against by having a separate emergency line, normally not under pressure, connected only to a hand pump or normally blocked off, or both. Connection between emergency and main lines may be established in various ways, the most usual being through a shuttle valve, one type of which is illustrated in *Fig.* 11. This illustration also shows a method for providing against failure, or omission to operate the selector, by having fluid from the return side of the cylinder released by a blocking valve automatically opened with pressure from the emergency line.

LIVE AND DEAD SYSTEMS

Noncontinuous and Continuous Systems: The name "noncontinuous systems" is given to systems which fulfill both of the following two conditions:

1. There is only one motor element (cylinder).
2. The pump is made to deliver only when the cylinder is to be operated.

All other types are termed "continuous systems." Condition 2 is fulfilled if the pump is hand-operated or if it is electrically operated but automatically switched off when the motion is completed. The reason for distinguishing between these two types of systems is that noncontinuous systems constitute as it were an elementary class wherein circuit problems are considerably simplified.

Noncontinuous Systems: The simplest system of this type is the well-known hand pump operated lifting jack *Fig.* 12. These are usually single-acting, a spring being provided sometimes to effect return in the absence of sufficient external loads. The cock is usually (as shown) biased towards the closed position and operation of the pump then causes the jack to extend. The jack will close if the cock is opened. Conveniently the cock may be actuated by a grip lever on the hand pump handle.

An ingenious variation of this type of system shown in *Fig.* 13, represents a hydraulic circuit for seat adjustment based on the use of an accumulator, which also serves as tank. The jack is retracted by operating the hand pump and a volume of oil equal to that displaced by piston rod is displaced into the accumulator, storing the corresponding amount of energy.

When the valve cock is opened, the accumulator pressure is applied to both sides of the piston, acting

Fig. 12—Left—Simplest type of noncontinuous system constituting a handpump operated lift

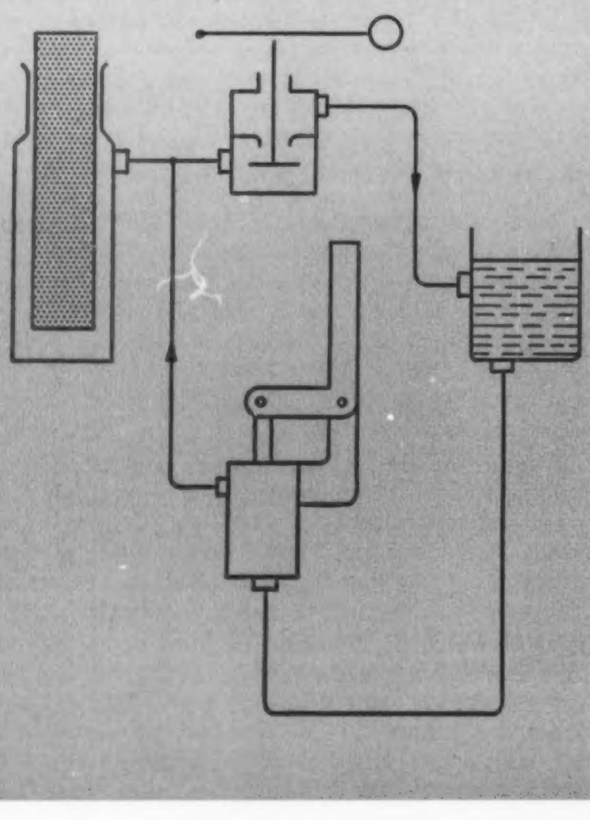
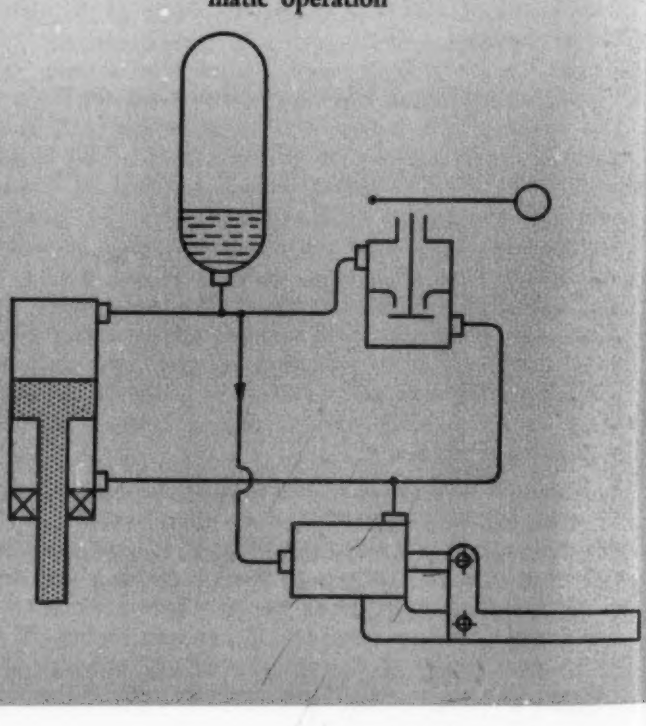


Fig. 13—Below—Variation of the system of *Fig.* 12, this design utilizes an accumulator for automatic operation



effectively on the area of the piston rod, and the cylinder extends.

In one type of hand remote control, a double-acting cylinder is moved in one direction or the other according to the sense in which the handwheel is turned (in this type of control, exact correspondence between handwheel and cylinder action is not expected). The handwheel operates a pump of the type which delivers in the same direction irrespective to the sense of rotation; reversal is achieved by a selector operated by some form of lost motion arrangement between the handwheel and pump, e.g., by a differential gear as shown in *Fig.* 14.

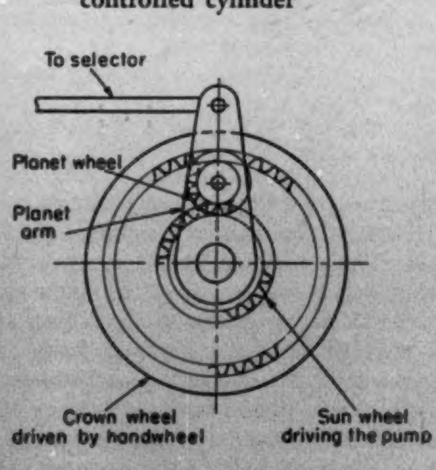
A circuit for operating pusher gear for loading bars into a furnace is shown in *Fig.* 15. The pump is driven by an electric motor, which is switched on to extend the cylinder. The pushbutton energizes a relay which switches on the motor through contact R_1 and holds on through contact R_2. The pump flow operates the valve V, connecting it to the cylinder and at the same time disconnecting the latter from tank. Before the cylinder reaches its stops, it operates a position switch which opens the holding line and releases the relay, thus switching off the pump motor. The valve V returns under the action of a spring, connecting the cylinder to tank and allowing it to return under the action of a spring or weight. A relief valve may be added to prevent mishap in case of failure of the limit switch to operate before the jack reaches its stops.

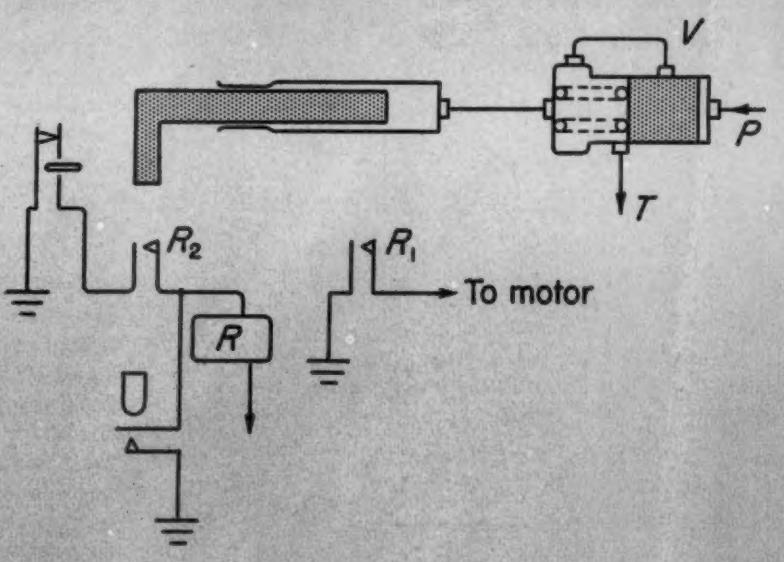
In circuits of this type it is of course also possible to cut out the motor by means of a pressure switch designed to operate when pressure rises due to the cylinder reaching its stops. This switch may take the form of a pair of contacts actuated by a spring-loaded plunger (or a Bourdon tube), the spring setting determining the operating pressure. A relief valve is then still necessary to prevent undue pressures being set up by motor inertia and other sources of delay. This relief valve must, of course, be set to a pressure slightly higher than the operating pressure of the switch. It seems sounder to use the relief valve as the single governing element and to actuate the switch from the outlet flow of the valve, as shown in *Fig.* 16. In this case the plunger is returned by a light spring.

In general, limit switches on hydraulic cylinders are not satisfactory if the cylinder is expected to reach its stops. Pressure or flow switches are to be preferred.

Fig. 17 shows an electrohydraulic system for actuating a double-acting cylinder. Here the pump is of the type which reverses flow when driven in the opposite sense. Reversal of cylinder motion is obtained by reversing the direction of the rotation of the motor. At the same time the selector is auto-

Fig. 14—Differential gear arrangement for operating hand remote controlled cylinder





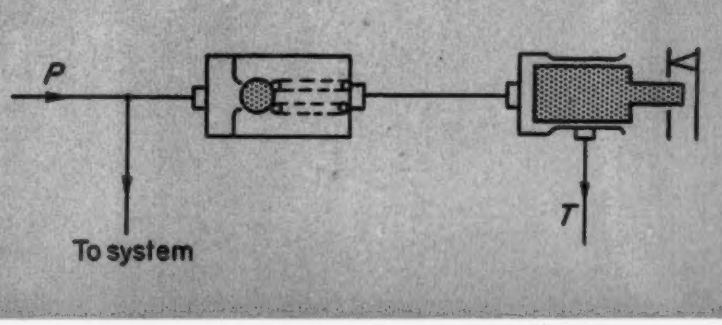


Fig. 15—Right, above—Furnace pusher gear circuit automatically powered by starting the motor

Fig. 16—Right—Motor cut-out scheme for a circuit of the type shown in *Fig.* 15

matically operated according to the direction of flow.

Continuous Systems: Continuous systems are the most widespread and also the most generalized form of hydraulic circuit. In continuous systems the required motion is determined by operation of hydraulic control gear, usually selectors, blocking valves (more rarely), and sequence valves (in automatic-sequence systems). This implies that hydraulic power is continuously available or must be made available when any motion is selected. In the latter case, means must be provided for turning the power on when motion is required, and of turning it off when it is completed. In the former case there arises the problem of disposing of the pump flow when no motion is required. In both cases, the problem will be referred to as "the idling problem."

The control problem and the idling problem may be said to be the general circuit problems of all hydraulic systems of the continuous type.

The Idling Problem: Generally, the idling problem arises with hydraulic systems of the continuous type having power-driven constant-displacement pumps. In such a system conditions arise, usually upon a cylinder completing its travel or a valve being closed, where no outlet would be offered to the pump delivery, unless a special path is opened to the flow under these particular conditions. A pump of this type is compelled to deliver at a given rate (apart from volumetric efficiency limitations) as long as it is driven. If the flow has no outlet, the pressure will rise until the efficiency of the pump drops to zero, part of the pump or circuit breaks, or the driving means stalls.

The expression, "idling condition" will be used to describe the condition where no part of the circuit is doing any work, including the case where load on one or more cylinders must be maintained when nominally at a standstill as, say, in plastic molding presses.

To deal with the idling condition, therefore, either the pump must be stopped or an outlet provided for the flow. In the case of a variable-delivery pump there is a third solution; reducing delivery to zero.

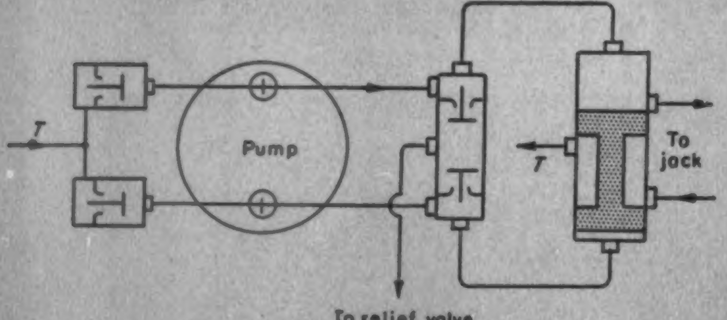
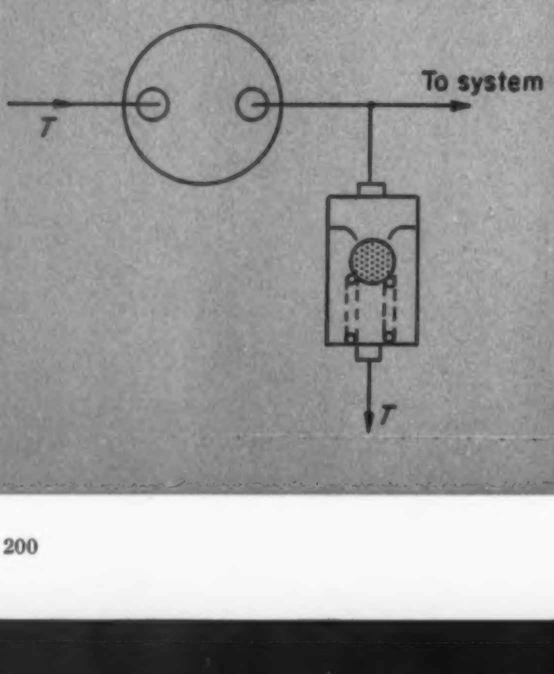


Fig. 17—Above—Electrohydraulic system in which cylinder reversal is obtained by motor reversal

Fig. 18—Below—Simplest solution for unloading a continuous system utilizing a relief valve



Power Dissipation and Unloading: Coupled with the idling problem is that of power dissipation or unloading. While no work is required of the pump, it will still consume full power if it has to pump fluid at full pressure in the idling condition. This condition occurs in the simplest solution to the problem, provision for idling by relief valve, *Fig.* 18. Power dissipation may be objectionable on the following grounds:

1. Excessive wear of the pump through constantly working at full load. This is often of minor importance, as many pumps are designed to work satisfactorily under these conditions.
2. Power consumption. This may be a factor of some importance in a number of cases.
3. Overheating of the oil. This is usually the most important consideration.

Relief-valve systems in which power dissipation is tolerated are certainly the simplest solution to the idling problem and the most satisfactory, provided none of the foregoing objections applies. This may be the case when the power of the pump is low and the heat generated is easily dissipated from the tank, without requiring excessive size, merely by relying on radiation and convection losses. It may also be the case in automatic machinery where the idling periods are short, provided there are adequate safeguards against leaving the pump running while the machine is not operating. In some automatic machines a fairly high degree of power dissipation is tolerated and air or water cooling of the oil is used.

In most systems, however, power dissipation is prevented either by stopping rotation of the pump or, more frequently, by-passing output to tank through a path of low resistance ("short-circuiting" or "unloading"). Both these solutions may be termed "unloading systems."

Live and Nonlive Systems: The term "live" will

be applied to systems in which pressure is maintained in at least part of the delivery main during idling. The simplest live system is one in which idling takes place by blowing off through a relief valve, *Fig.* 18. In other types of live systems to be described in the following paragraphs, the pump is unloaded while pressure is maintained, this result being achieved by some automatic regulating system.

With a very few minor reservations, live systems provide a general solution to the idling problem. Any number of services may be supplied with such a system and no special consideration need be given to the interrelation between the operation of any service and unloading. In other words, one single regulating device responds only to the pressure in the delivery main. In nonlive or dead systems, when a selector is operated there would be no pressure available unless specific provision is made simultaneously to put the system on load again by restarting the pump or cutting off the unloading path, as the case may be. This provision may take various forms according to the requirements of the machine in question, and it may be said that as far as dead systems go there is no perfectly general solution.

Live systems still have their limitations however. In many cases the presence of maintained pressure is disliked and for many machines dead systems can be devised which are simpler and less delicate than live ones. Indeed for self-contained hydraulic machines and for motor vehicle hydraulics, nonlive systems are the rule rather than the exception.

Two-Pump Systems (Partial Unloading): In a live system only a small output is required to maintain pressure when idling—enough to deal with leakage losses and thermal contraction effects. Hence, the function of maintaining pressure can be vested in a small pump discharging through its own relief valve during idling periods, while the main pump is automatically unloaded during these periods. The power of the small pump is easily kept low enough to prevent appreciable heating of the oil.

The basic circuit is shown in *Fig.* 19. The loaded and unloaded parts of the circuit are separated by a check valve. When the circuit pressure reaches a predetermined value, the unloading valve begins to function as a relief valve for the main pump. Due to the small pump, however, the circuit pressure rises further and a slight increase will cause the unloading valve to open fully, thereby unloading the main pump. The final maintained pressure is determined by the setting of the relief valve, which must be slightly higher than that of the unloading valve. For a really neat installation the small pump should be incorporated with the main pump.

This system is qualitatively similar to the "quick-approach" type of two-pump system, but the functions are quite different, the object being to prevent power *dissipation* and not to reduce peak power requirements. Quantitatively there is an important difference, because in the quick-approach two-pump system the setting of the relief valve (or more generally the operating pressure of the smaller pump) is much higher than (seldom less than twice) the setting of the unloading valve, and the size of the smaller pump is governed by different considerations.

Automatic Cut-Out Systems: Systems of this type are fundamentally similar to the two-pump system just described but the function of the small pump is performed by a reserve of fluid under pressure. This reserve must be provided by an accumulator or its equivalent. With this system, power dissipation is strictly limited to the requirements of leakage and thermal contraction make-up.

This type of circuit is in current use in old-fashioned water-hydraulic systems, usually so proportioned that supply from the accumulator is available at all times. Systems of this type frequently use a weight-loaded accumulator, and in this case the cut-

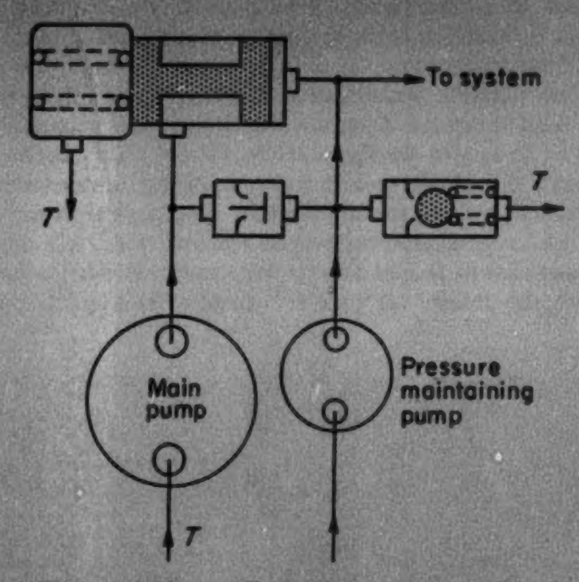
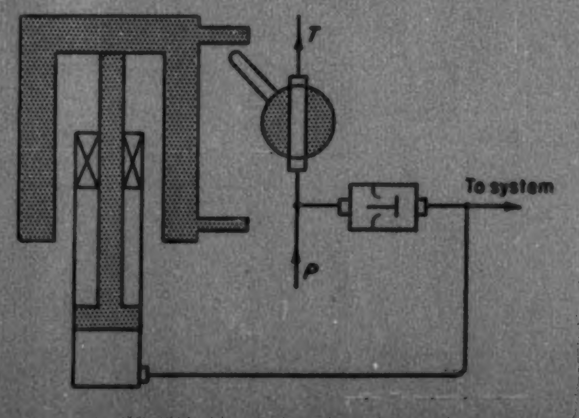


Fig. 19—Above—Basic live system employing two pumps and unloading valve to cut out the main pump

Fig. 20—Below—Weight-loaded accumulator system with automatic cut-out valve



out device may consist of a stop valve fitted with a snap-action mechanism. The cut-out is operated by the accumulator weight and opens when the latter reaches a position near the top of its travel, closes when the weight has traveled down a pre-determined amount. The circuit is shown in *Fig*. 20. The check valve isolates the loaded and unloaded parts of the circuit.

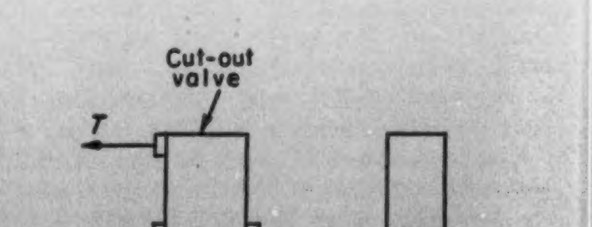
In modern systems the cut-out valve is usually pressure sensitive instead of mechanically dependent on accumulator position. Pressure-sensitive valves open at a certain pressure or cut-out setting, and close again at a somewhat lower pressure or cut-in setting. A basic system of this type is shown in *Fig*. 21. Again the cut-out valve embodies a check valve which separates the loaded and unloaded parts of the circuit. The valve responds to pressure at connection *C*. When this reaches a certain value p_1, a connection is established between ports *P* and *T*, and the pump is unloaded, the action being such that when the pressure p_1 is reached the valve opens fully. When the pressure drops to a value p_2 less than p_1 by a definite margin, the valve closes again, preferably suddenly.

One possible construction for a cut-out valve specifically designed to give instability throughout its travel is shown in *Fig*. 22. The spool *S* is urged towards the closed position by a spring and a small piston *B*, which are opposed by the larger piston *A*. When the cut-out pressure is reached, piston *A* overcomes piston *B* and the spring, and the spool moves left. As soon as port *T* begins to open, part of the pump flow goes to tank through the low resistance *R* thereby creating a slight pressure at connection *C* which acts on the spool (the pressure also acts against the piston *A*, the area of which must be less than that of the spool to give a net increase). The load to the left on spool thus increases and the valve opens further; pressure at port *P* drops, and the load exerted by piston *B* decreases. The net result is that at the cut-out pressure the valve will snap open.

When the valve is open and the pressure in the loaded part of the circuit (acting at port *D*) drops to the cut-in value, the valve begins to move to the right until port *T* is partially closed, thus causing a rise in pressure at port *P* which causes piston *B* to exert additional load to the right. Here again an unstable condition is reached (and furthermore is reached before substantial resistance is offered to the pump flow) and when the valve is fully closed, pressure at port *C* drops to zero. The resistance *R* causes a slight pressure to be maintained on the pump during the unloaded condition; slight residual pressures are often beneficial, in any event, because high-pressure pumps do not work well when the output pressure is practically zero.

In the circuit of *Fig*. 21 the function of the accumulator is twofold: (1) It supplies the small volume of fluid required to open the valve; and (2) it prevents overfrequent cutting in, the inevitable result in a low-capacity circuit in which the slightest loss of fluid volume would result in a sharp drop of pressure. If the accumulator has no other function

Fig. 21—Right—Basic accumulator system utilizing a pressure-sensitive control



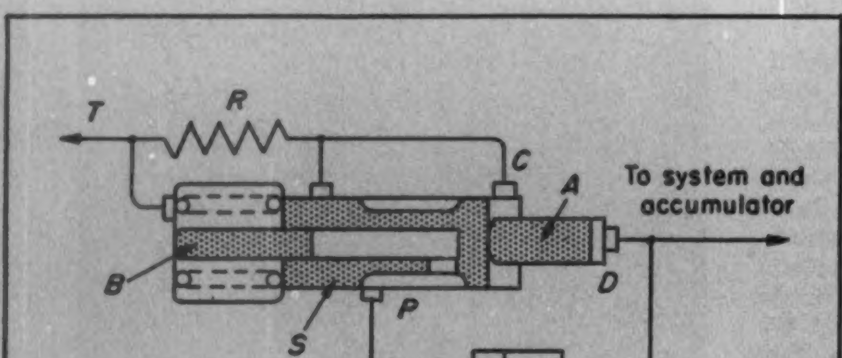


Fig. 22—Left—Design for a pressure-sensitive automatic cut-out valve

to perform, it can be small in volume, in some cases small enough to allow the use of spring loading. Flexible hoses have surprisingly high elasticity, and a few feet of hose may be adequate as an accumulator in some cases.

Large volumes of fluid have appreciable elasticity if the working pressure is comparatively high and, if the cut-out valve is in *permanent* communication with such a volume, a separate accumulator may be avoided. Such a volume of fluid is available if the delivery line is in communication with a large cylinder, although it is cut off during crossover of the selector from one extreme position to the other and, in such cases, it seems desirable to provide a relief valve set to a pressure somewhat higher than the cut-out pressure. The cylinder capacity cannot be utilized if the selector is to be left in neutral. A permanently available volume of fluid may be provided by a pressure bottle (filled with oil), a very convenient form, in many cases, of low capacity accumulator.

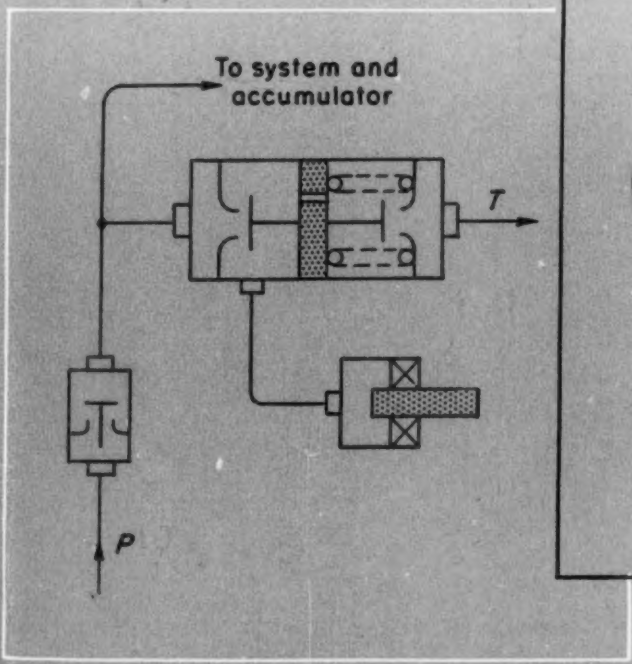
The margin between cut-out and cut-in pressure cannot be small in practice; 10 per cent of the cut-out pressure would seem a reasonable minimum figure. Furthermore, it is usually unsound to allow pressure during any operation to exceed the *cut-in* value, as otherwise if the operation is stopped under peak load it cannot be resumed since the pump will not cut in. Hence, the peak rating of the system based on the cut-out pressure—reached only after completion of an operation—must be, say, at least 15 per cent higher than the actual peak power requirement of the machine to be operated. This is a slight drawback of this type of system. It may, however, be partly or even wholly offset by pressure losses in the pipes and valves if these losses are at all large.

Live Systems with "Drive" Unloading: In the case of drive from an engine, unloading may be carried out by declutching the pump; in the case of electric motor drive, by switching off the motor. In both, if the system has some capacity (accumulator or equivalent) and if unloading is effected by some automatic device which cuts out at a certain pressure and cuts in at a somewhat lower pressure, a live system is obtained.

Neither of the foregoing systems is well suited to applications which call for cutting in and out at frequent intervals, nor indeed is there much to favor them for any application, as most pumps last better when running continuously with long periods under light load (as they would do with by-pass unloading) than when they are frequently started and stopped. The starting is particularly brutal with the declutching system. If power is required only at infrequent intervals there is usually little to be said in favor of a live system in any event.

An automatic system suitable for either case is shown in *Fig.* 23. The cut-out device is a pilot valve of the seating type, spring loaded to the left-hand port. The opposite port is somewhat larger, the ratio of the two port areas determining the ratio of the cut-out to the cut-in pressure (with some correction for spring rate). When the cut-out pressure

Fig. 23—Below—Automatic system suitable for drive unloading or declutching



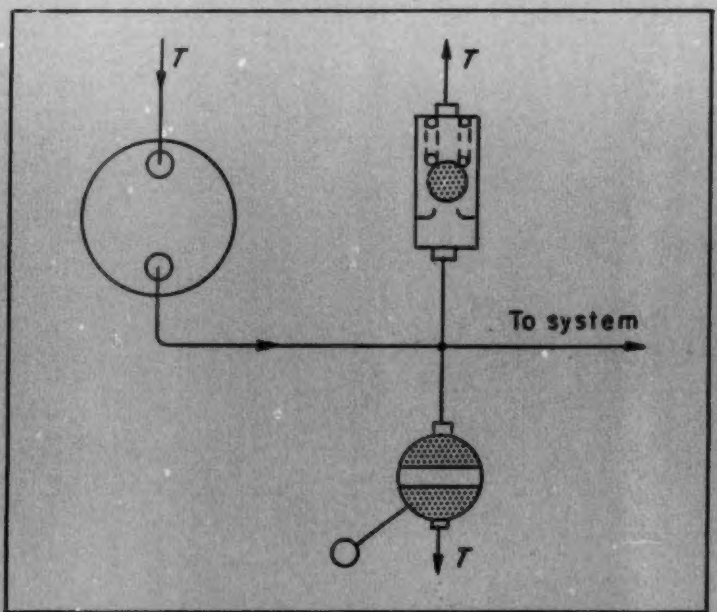


Fig. 24 — Above — Elementary system for short-circuit unloading

is reached, the valve begins to move and the pressure temporarily acts on the full piston area causing the valve to seat rapidly on its right-hand port, where it remains until pressure has dropped to the cut-in value. On cutting out, pressure is admitted to the cylinder, which may disengage a clutch or break a pair of contacts controlling a motor. When the valve resets, the cylinder is connected to tank and the clutch is engaged or the motor contacts are closed. Here again, a check valve isolates the loaded from the unloaded parts of the circuit, as otherwise pressure could leak away through the pump or cause the latter to turn backwards. The check valve can be dispensed with if the pump has automatic seating valves.

The circuit in *Fig.* 23 is also applicable to by-pass unloading, as the cylinder could equally well be the operating piston of an unloading valve. Actually the principle of all unloaded live systems is essentially the same.

Dead Systems: In nonlive or dead systems, since no pressure is maintained, there cannot be a pressure-sensitive device such as a cut-out valve to put the pump back on load when its output is required. Hence, in a dead system, the operation of a selector or any other device with a similar function (e.g., a switch in an electrically controlled system) must be coupled with the operation of closing the short-circuit path or switching on the motor, where applicable.

With electric motor drive it is always possible to switch off the motor after any operation has been performed and this method of unloading is adequate for certain applications in which the services of the pump are required only at comparatively rare intervals, e.g., in furnace door lifting gear.

With short-circuit unloading the most elementary system is shown in *Fig.* 24. After any operation is completed, the blocking valve is opened by the operator, who must close it again when the next operation is to be performed. The danger of the blocking valve being left closed is avoided by making its opening automatic. One possible way of doing this is illustrated in *Fig.* 25. The blocking valve is hand closed against a spring, and kept closed by a latch. The latter is withdrawn when pressure is sufficiently high to overcome the spring (as shown) or, better, by

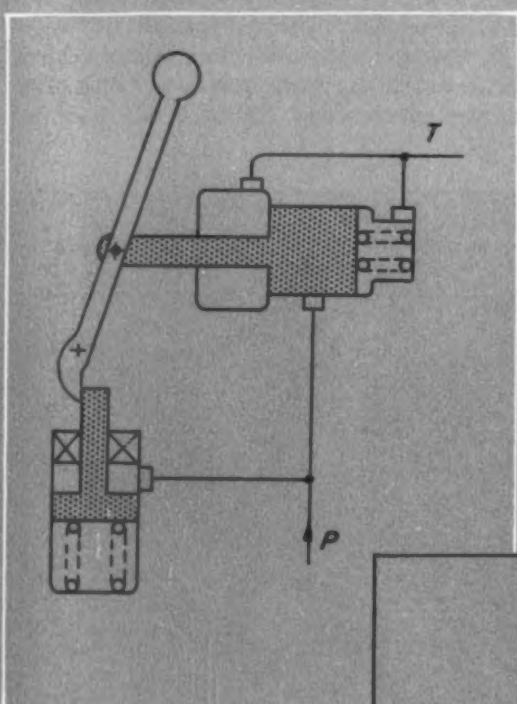


Fig. 25—Left — Automatic blocking valve for short-circuit unloading system

Fig. 26 — Below — Slide valve with spring bias utilized for unloading

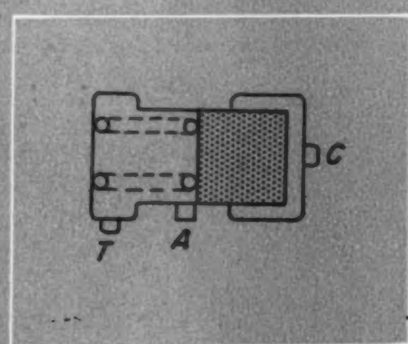
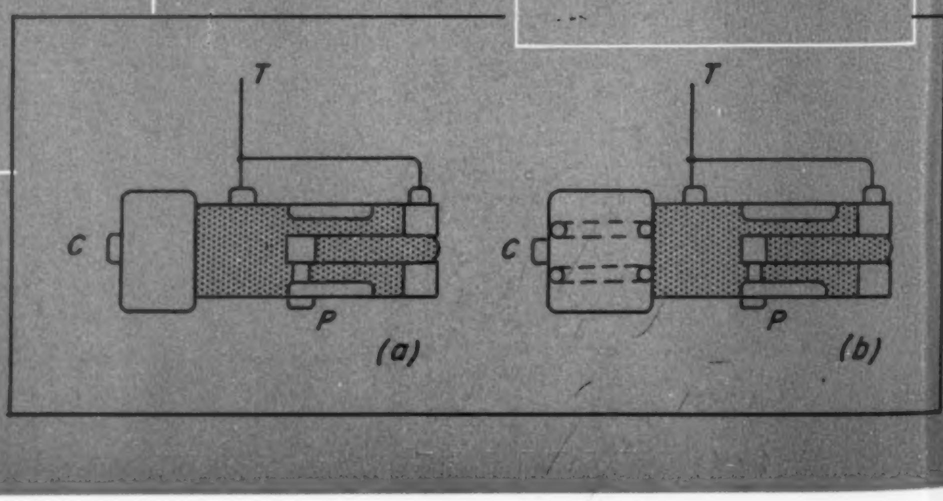


Fig. 27—Below—Unloading valve designed to open and close with presure, *a*, and with spring bias to provide slight residual pressure in unloaded condition, *b*



being operated by flow from the relief valve outlet, thus automatically unloading the pump at the completion of any operation. Equivalent electric systems are easily devised in which the motor is automatically switched off.

The system of *Fig.* 24 is perfectly sound, but involves the usually undesirable additional maneuver (as compared with fully automatic systems) of closing the blocking valve at each operation. To avoid this problem, it is possible to arrange the circuit so as to cause the unloading path to be cut off in the following ways:

1. If the system operates one service, or a number of independent services, the circuit can be devised to cause the operation of *any* selector (or equivalent element) to cut off the by-pass path (or restart the driving motor if applicable) and provide for automatic reopening on completion of the operation. With hand-operated selectors this leads to the important class of open-center systems and allied types of circuits. All solutions of this type may be termed "coupled systems" since the operation "selection" is *coupled* with the operation "putting back on load."
2. If the system operates one single machine which has to perform certain operations in a definite sequence, the by-pass may be automatically closed on operating the valve or switch which initiates that sequence, and automatically opened when the sequence is completed. Circuits of that type concerned with the general problem of automatic sequence may be termed "sequence unloading."

Short-Circuit Unloading: In the case of pumps having automatic check valves, it is possible to effect unloading by positively lifting the suction valves off their seats, thus avoiding a separate unloading valve. This is an old-fashioned solution which in practice has little to commend it.

If unloading takes place through a pressure-closed unloading valve, the latter cannot be closed unless some residual pressure is present in the unloaded condition. Although the unloading valve is of some fully balanced type, frictional forces are still present, and a definite load is required to open it. Even in a balanced slide valve, the friction load is roughly proportional to the pressure at blocked connections, which may be (indeed usually is at the instant when

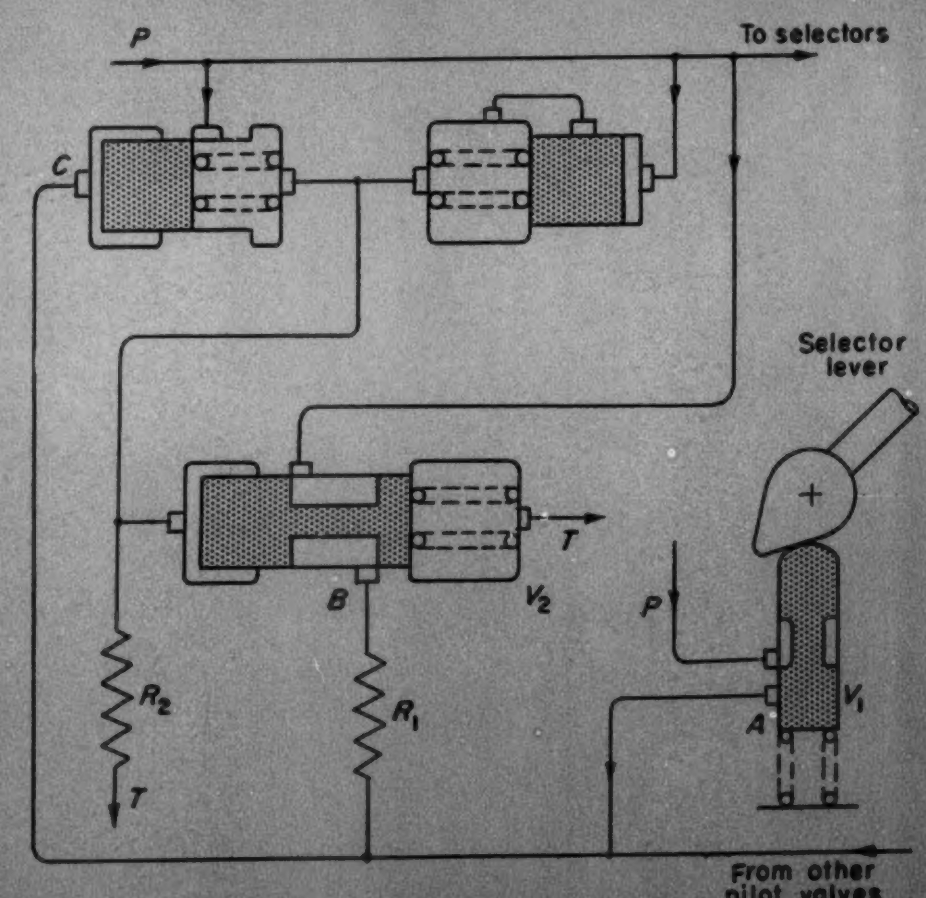


Fig. 28—Circuit design with separate relief and unloading valves

the valve has to open) the full system pressure and the load required to operate the valve may be quite appreciable. In the slide valve of *Fig.* 26, the spring must supply this load, and the residual pressure in the unloaded condition (when the valve is open) must be sufficient to overcome the spring load. This may call for values of perhaps 5 or even 10 per cent of the maximum system pressure, which may easily mean that in the unloaded condition, power dissipation would still be excessive.

This difficulty may be avoided by using an unloading valve which is opened as well as closed by pressure, the pressure for opening acting on a relatively small area. An example of a slide valve using this principle is shown in *Fig. 27a*. If a weak spring is added, *Fig. 27b*, the valve will cause a slight residual pressure to be maintained in the unloaded condition when it acts as a low-pressure relief valve. The pressures required to close this type of valve can be kept to extremely low figures. *Fig.* 27 shows the valves in the closed position, i.e., pressure is assumed to be applied at *C*.

Another important consideration must be borne in mind in connection with pilot control of unloading. The unloading valve is usually opened by putting the "control" connection to tank through a pilot valve. The unloading valve is designed to close under low pressure and, hence, to enable it to open the "tank" connection of the pilot valve must be at atmospheric pressure, or very nearly so. This condition is not necessarily obtained if the tank line from the pilot valve branches into a return line having substantial flow through it, because appreciable pressure difference would exist between the branch and atmosphere.

In the circuit of *Fig.* 28 is shown an unloading valve and a relief valve as separate items. In practice, a single valve is often made to do the work of both—a highly desirable simplification. One valve of this type is shown in *Fig.* 6. When the control connection is put to tank, the controlling spring expands to its free length (or until its load is taken by some cage arrangement) and, in effect, the relief valve is regulated down to zero pressure (a weak spring may be provided on top of the upper piston to maintain slight residual pressure). When pressure is applied to the control connection, the upper piston (which is larger than the spool) travels down to a stop (when pressure becomes high enough—it

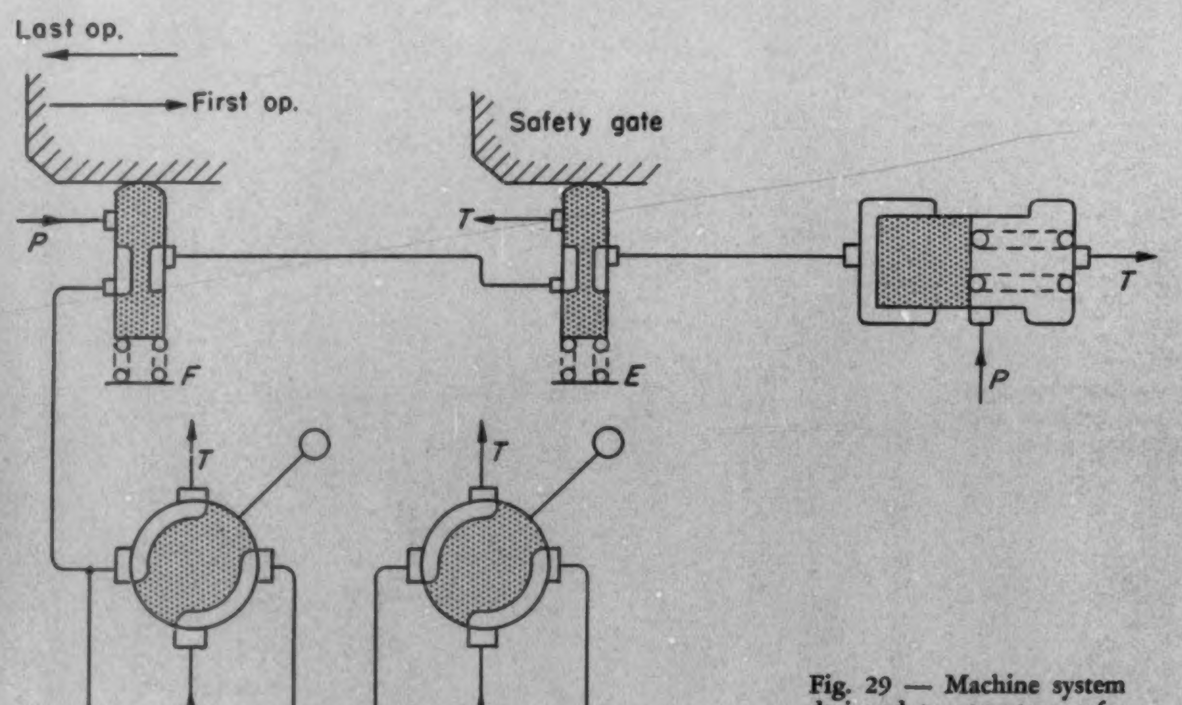


Fig. 29 — Machine system designed to operate a safety gate by pilot circuit

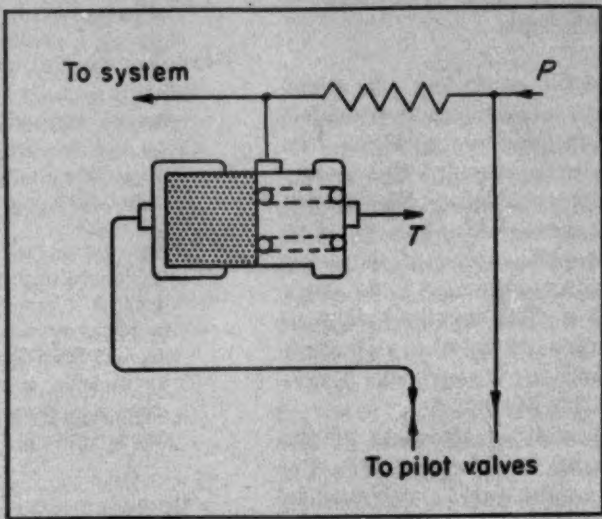
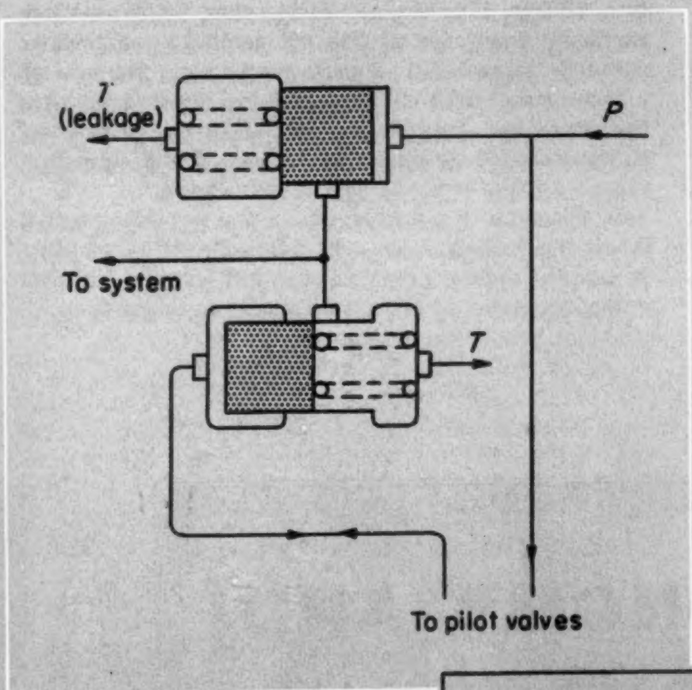


Fig. 30—Right—Circuit example showing method of creating residual pressure to close the unloader without movement of any machine elements

Fig. 32—Below—Safety circuit arrangement utilizing a low-pressure blow-off valve to prevent residual pressure from reaching the main circuit



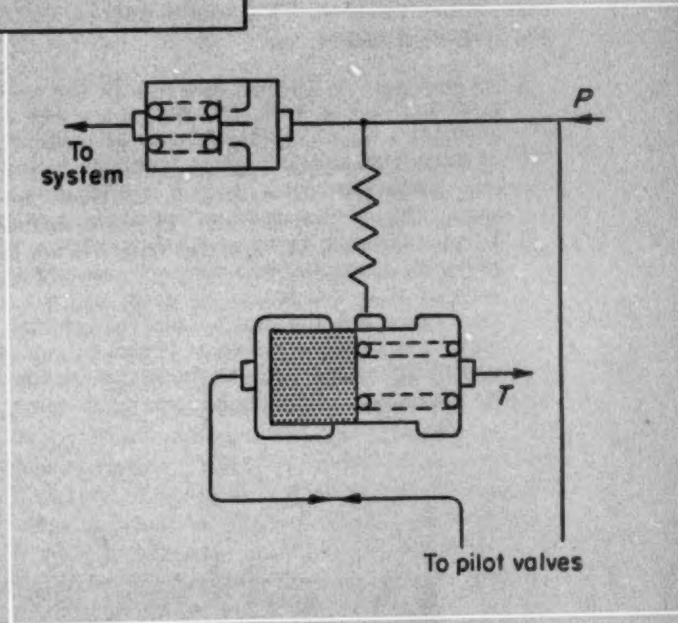


Fig. 31—Above—Circuit alternate for that of *Fig.* 30 employing a pressure sequence valve

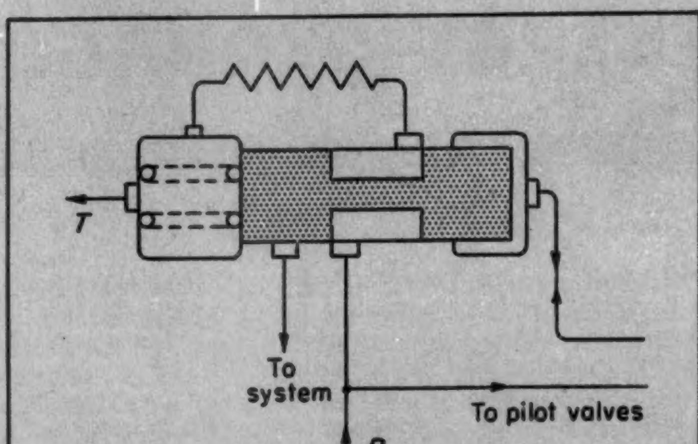


Fig. 33—Left—Three-way valve unloading circuit designed to cut off the main circuit

is immaterial whether or not the piston completes full travel immediately) and the spring is compressed to give the desired blowing off load.

Operation of Safety Guards or Gates: In many presses and other machines a safety gate is provided, the arrangement being such that the machine cannot function until the gate is closed. The safety gate may operate a switch controlling the driving motor, the switch being open until the gate is shut. In most cases, however, stopping the driving motor is undesirable and the usual arrangement is to cause the safety gate to operate a pilot valve which controls an unloading valve, the latter also unloading the system under other conditions if required. A system of this type is illustrated in *Fig.* 29.

In such cases, the main moving elements of the machine must be incapable of being operated in the unloaded condition. This object can be achieved in the following ways:

1. By making the residual pressure in the unloaded condition so low that it cannot move any of the elements of the machine. Often, the initial motion of these elements is against frictional forces only and fairly low pressures are sufficient to start them. Thus, the residual pressure maintained by an unloading valve of the type of *Fig.* 27 can easily be excessive. The residual pressure (necessary to close the unloading valve and sometimes also for other purposes) must be generated in other ways. One example is shown in the circuit of *Fig.* 30, where a residual pressure is maintained by a resistance, but reaches the pilot valves only and not the main system. Instead of the resistance, use could also be made of a lightly loaded blow-off valve which would have the advantage of giving a pressure independent of flow, if the latter can vary. In practice, a check valve could be used with springs perhaps slightly more powerful than would normally be fitted. A third possibility is to use a pressure-sequence valve, *Fig.* 31, which has the advantage that there is no power dissipation when the system pressure exceeds the valve setting.
2. By preventing residual pressure from reaching the main circuit. This can again be done by interposing a low-pressure blow-off valve, *Fig.* 32, or a pressure-sequence valve set to a value slightly higher than the residual pressure. Another method is to use a three-way valve for unloading in such a manner as to cut off the main circuit when the by-pass is opened, *Fig.* 33.

In the circuits of *Figs.* 30 and 31, the unloading valve could be of the type of *Fig.* 27*a*. In the circuit of *Fig.* 33, the unloading valve could be of the type of *Fig.* 27*b*, the resistance then being omitted. Similarly the valve of *Fig.* 33 could be modified to maintain some residual pressure without the use of a resistance. The circuits of *Figs.* 32 and 33 give the maximum of safety, since the system is immune to pressure set up by the resistance of the unloading valve and the pipes in the by-pass path.

In the case of systems where the unloading valve is required for safety purposes only, it is possible to use the safety gate to actuate directly a full size unloading valve of the mechanically operated type.

POWER ECONOMY AND ENERGY STORAGE

The Approach Problem: In many hydraulically operated machines the working stroke comprises an initial portion during which load is extremely small or even negative. This condition may arise, for instance, in presses where the stroke is determined by the space required to insert and remove the work (or similar considerations), in retractable aircraft undercarriages during the extension stroke, in gripping devices, etc. Indeed, conditions of this type are the rule rather than the exception.

The usual requirement is the completion of the full stroke within a given time which can be apportioned as convenient between the loaded and unloaded periods. If, however, a simple cylinder is to be moved by a constant-delivery pump driven at constant speed, the speed of the cylinder will be practically independent of load and, hence, the times for the loaded and unloaded portions of the stroke will be proportional to their respective lengths. Under such conditions it is easily seen that the peak power required is much

larger than that necessary when speed of movement varies inversely as the load in such a way as to give the same time for the complete stroke.

Power economy is always desirable, not only as an end in itself, but also to reduce the size and therefore cost of the pump and control gear, and to minimize heat dissipation. In many presses and other machines, however, power economy is absolutely essential if the machine is to be at all commercially practicable. Hence, in many applications means of moving the machine at higher speeds under low loads must be sought in order to even out the power curve and reduce the peak insofar as possible. In this area only *automatic* means of varying speed will be considered.

Load-Stroke Curves and Power Calculation: *Fig.* 34*a* shows a typical load-stroke curve applicable to many kinds of presses and also to power brakes for vehicles. Here the approach and loaded portions are clearly distinguishable and during the former the load, usually entirely or almost entirely due to friction, is very low. At *b* in *Fig.* 34 is shown a typical load-stroke curve for an aircraft landing gear retraction cylinder. Here there is no clearly defined approach period and most of the quick-approach systems discussed herein are not applicable to this type of curve. *Fig.* 34*c* shows a curve applicable to down-stroking presses, in which the weight of the table is higher than frictional forces, resulting in a negative load during approach. In that for an aircraft wing unfolding cylinder, *Fig.* 34*d*, again there is a negative load at the beginning.

For any of these curves the average power required is given by the integral of the load-stroke curve divided by the time required for the stroke. If the hydraulic system is so arranged that the power is constant, this average is also the peak power. If cylinder speed is constant, the peak power is the peak load times the stroke, divided by the time for total travel.

Thus, if the time for the total stroke is kept fixed:

$$\frac{\text{Average power}}{\text{Peak power}} = \frac{\text{Area under curve}}{\text{Area enclosed by maximum co-ordinates}} \quad \ldots (9)$$

In the case of curves of type *b* of *Fig.* 34. this ratio will be called geometric efficiency.

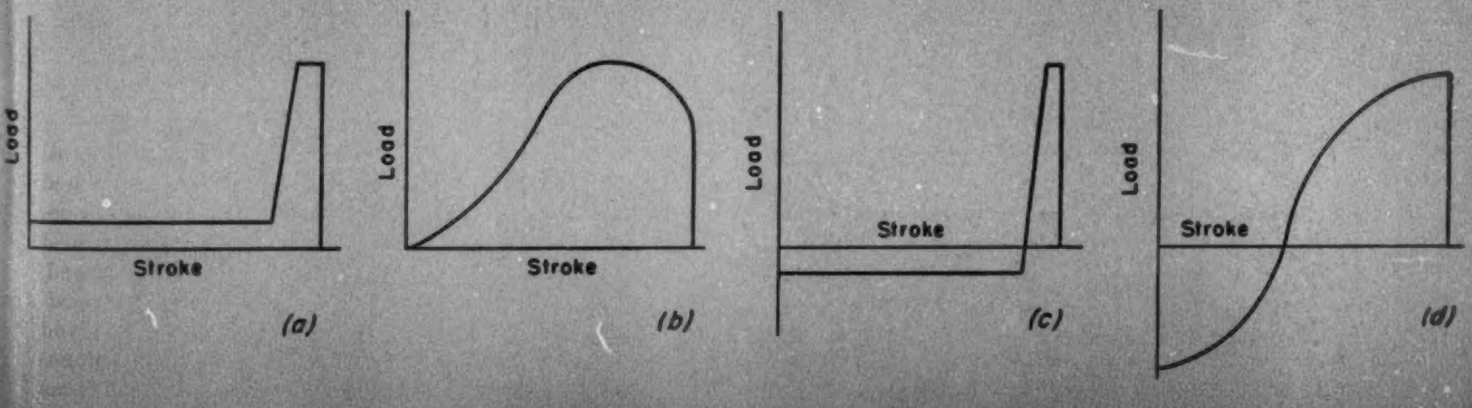
With load-stroke curves of type *a, Fig.* 34, the approach load is often negligible. It is often most convenient to calculate peak power by trying a given time for the loaded portion and then endeavor to find the highest practicable approach speed. It is often impracticable to use full power during approach, as the resulting speed would be excessive.

A similar procedure would apply in the case of curves of type *c* of *Fig.* 34. The procedure may be applicable even if the loaded portion of the stroke is nominally zero as in many plastic molding presses, printing machines, gripping devices, etc. In such cases the power required for the loaded portion can still be quite high because work is still done in deflecting the machine structure and in compressing oil in the pipes and cylinders. The power required can be calculated if a definite time is allowed for the load to build up to maximum. Work done in compressing the oil may be quite high with large presses in which the volume of oil in the cylinder may be considerable. In high-speed machines the volume of oil in the pipes may be the most important factor.

Solutions to the Approach Problem: If the pump is driven by a series wound dc electric motor (or any other motor of similar characteristics), the latter will automatically ensure higher speeds under smaller loads. This type of solution is of little interest except in a few special cases, particularly since the operating speeds of most pumps are limited.

Another solution, variable-delivery pumps, results in economy of power but not in size and cost of pumping gear. Variable-delivery pumps are perhaps the

Fig. 34—Typical load-stroke curves for various systems. That for presses is shown at *a*, for aircraft landing gear at *b*, for down-stroke presses at *c*, and aircraft wing unfolder at *d*



only efficient solution for machines in which the load is subject to variation over a wide range.

Further consideration of this problem involves the following basic quick-approach systems:

1. Twin-pump systems.
2. Variable-area systems.
3. Prefilling systems.
4. Accumulator and intensifier systems.

Twin-Pump Systems: Systems of this kind are probably the most widely used solution for machines of the press type. Among others, they have the advantage of generality, i.e., with a twin-pump system quick approach may be obtained for any number of operations carried out by different cylinders without further complication. In particular, quick return is automatically obtained if the return load is low.

A basic system is shown in *Fig.* 35. There is one large-delivery low-pressure pump *A*, and a small-delivery high-pressure pump *B*. At the end of the approach period pressure rises until at a specified pressure the valve *C* operates, unloading the pump *A* while the pump *B* continues to supply power. The valve *C* is set to open at a pressure somewhat higher than the maximum value required during the approach period. In practice there may be no gain in setting the operating pressure of the valve at a value much lower than

$$p_1 = \frac{p_2 q_2}{q_1 + q_2} \qquad (10)$$

where p_2 = maximum pressure during the loaded portion, psi; q_2 = output of the small pump, gpm; and q_1 = output of the large pump, gpm. The system of *Fig.* 35 is qualitatively identical with that of *Fig.* 19, but fulfills quite a different purpose and is quantitatively different.

It is frequently the practice to build twin pump sets for this type of system in a single unit, with one driving shaft and often with the unloading valve incorporated in the unit. Unloading of the large pump may also be controlled directly according to the position of the machine although it is desirable to retain an overriding pressure control as a safety measure. Position control may be desirable in such operations as die closing in molding or die casting machines, the object being to prevent the dies being closed at too high a speed and creating severe impact loads. It is then necessary to unload the large pump before the end of the low-load portion of the stroke. A system for a machine of this type is shown in *Fig.* 36. To avoid a multiplicity of valves, the unloading valve *A* is made double-acting and will open when pressure

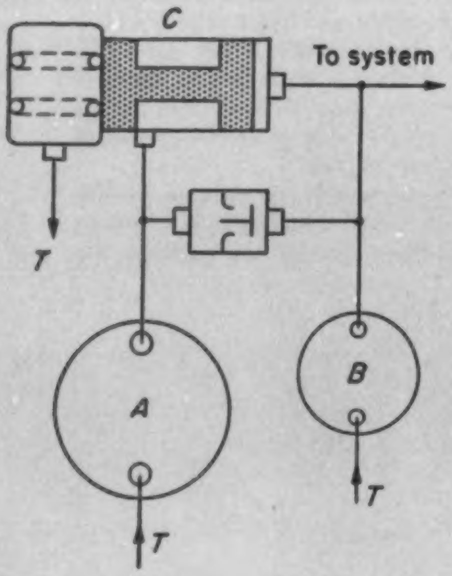


Fig. 35—Above—Basic twin-pump quick-approach system

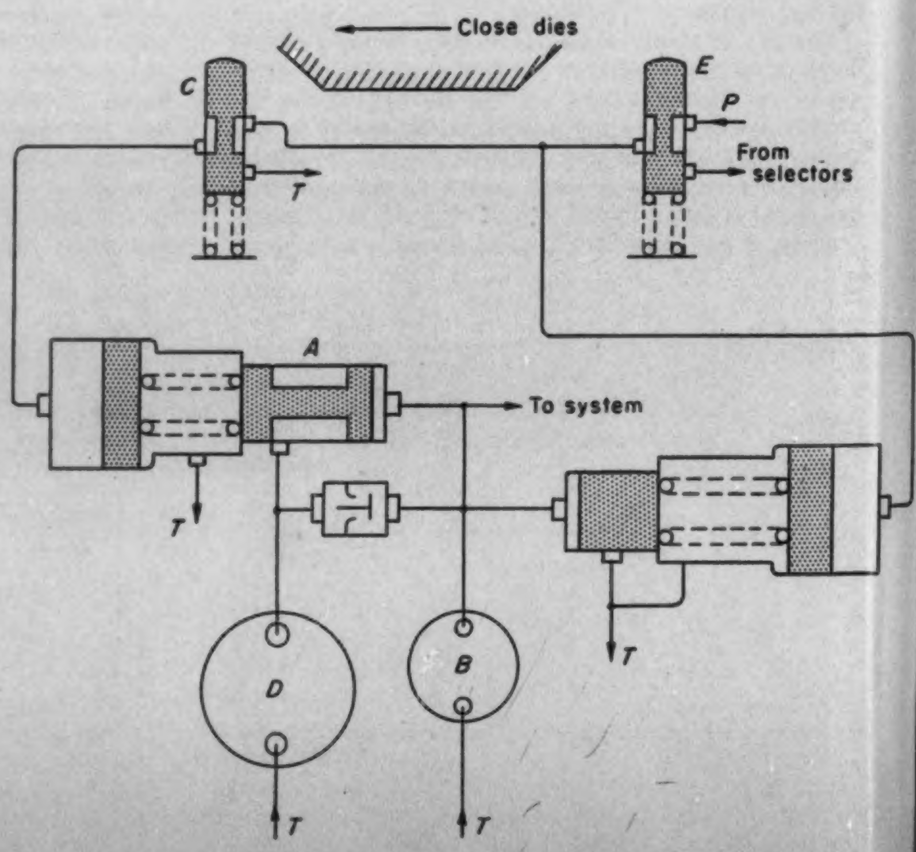


Fig. 36—Right—Circuit for a molding or casting machine featuring position control to effect shockless closing of the dies

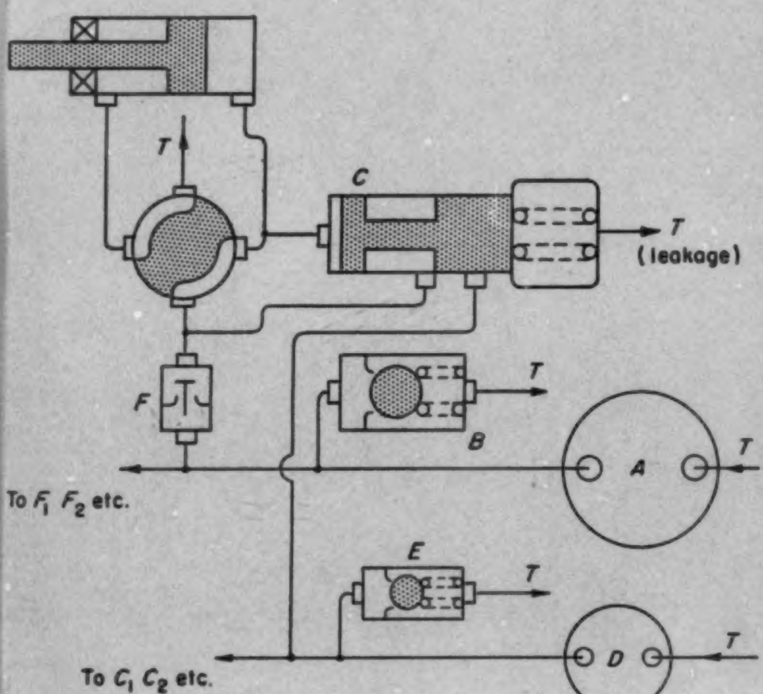
reaches the setting value, or when the top control connection is put to tank, thus releasing the load on the control spring. This type of valve should be compared with the double-acting relief valve shown in *Fig.* 6 and also used in the present circuit for the smaller pump *B*. Just before the dies are closed, the mechanically operated pilot valve *C* unloads the large pump *D* and, when the dies are fully open the pilot valve *E* unloads both pumps.

A two-pump system may be considered as an approximation to a variable-delivery pump. For load-stroke curves of types *a* and *c* of *Fig.* 34, it is probably more efficient than a variable-delivery pump which tends to have poor efficiency at small outputs. Two constant-delivery pumps are often less costly than one variable-delivery pump for the same purpose, even when allowance is made for the additional valve required by the former.

Where a speed ratio of 2:1 between approach and loaded portions is acceptable, it is possible to use two identical pumps working in parallel for approach and in series for the loaded portion of the stroke during which they function as a two-stage pump. The main point of such a system arises in cases where the required pressures are higher than those which can be reached with a given type of pump in a single stage. In any event multistage pumps present special problems and the pumps must be specially designed for the application.

Fig. 37 shows a variation of the two-pump system, adapted to operate a battery of independently controlled presses which remain under load for consider-

Fig. 37—Variation of the two-pump system devised to operate a battery of independently controlled presses



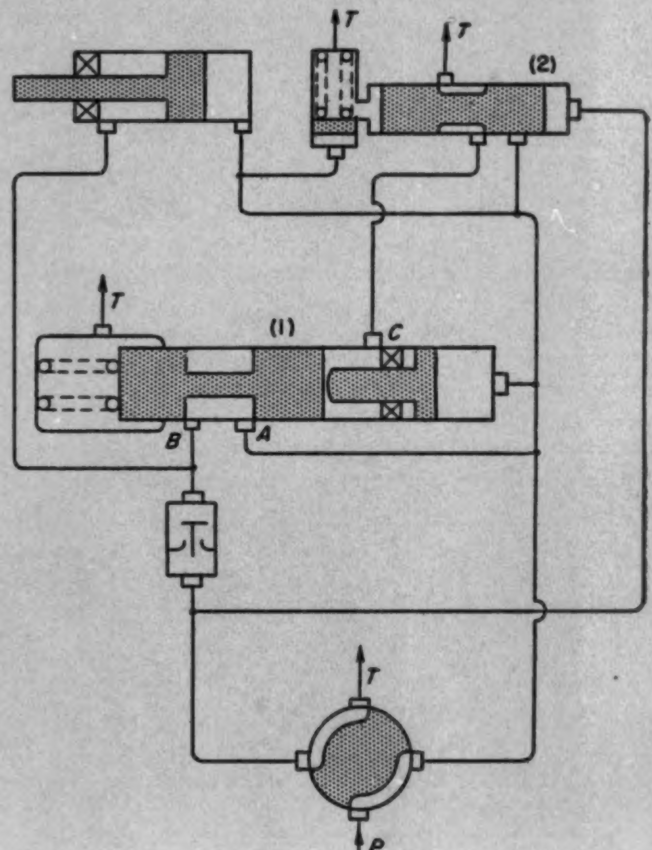


Fig. 38—Right, above—System designed to operate on differential piston areas

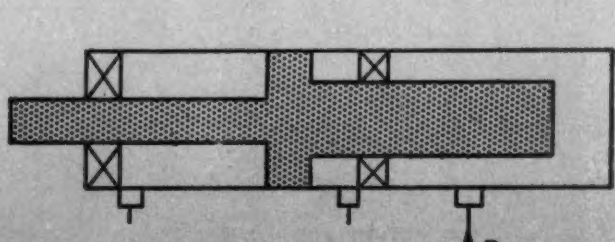


Fig. 39—Right—Three-chamber cylinder employed for quick-return motion

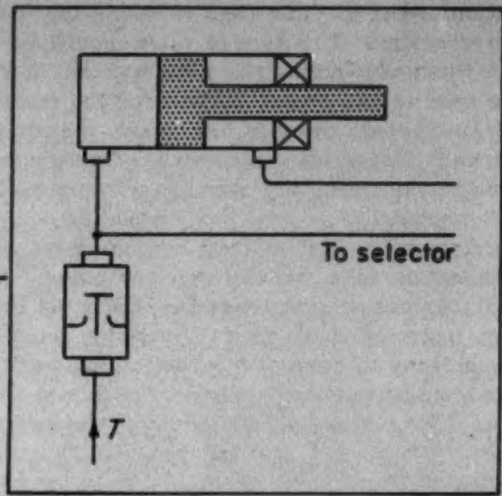


Fig. 40—Right—Negative-load cylinder arrangement designed to prefill from a tank under extension stroke

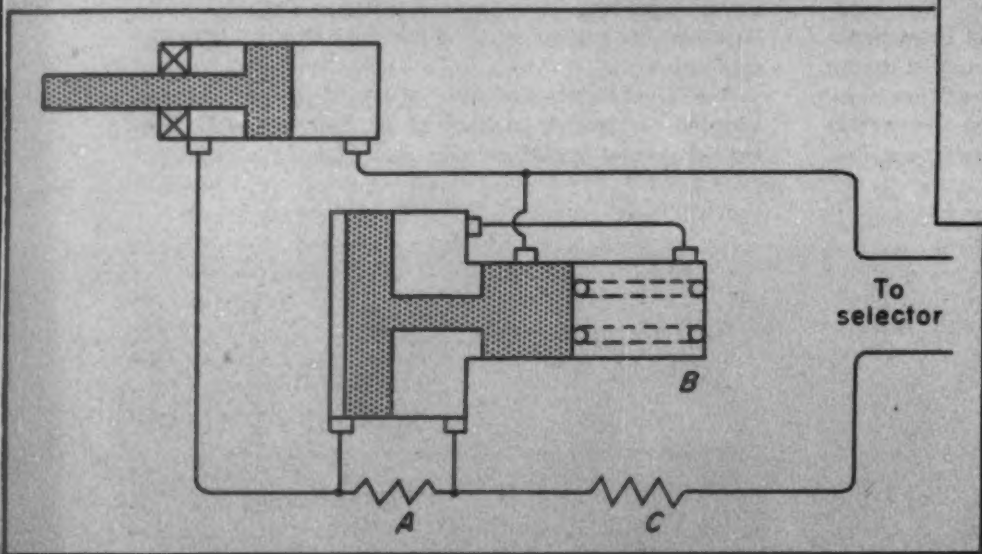


Fig. 41—Below—System designed for prefilling from the opposite side under negative load

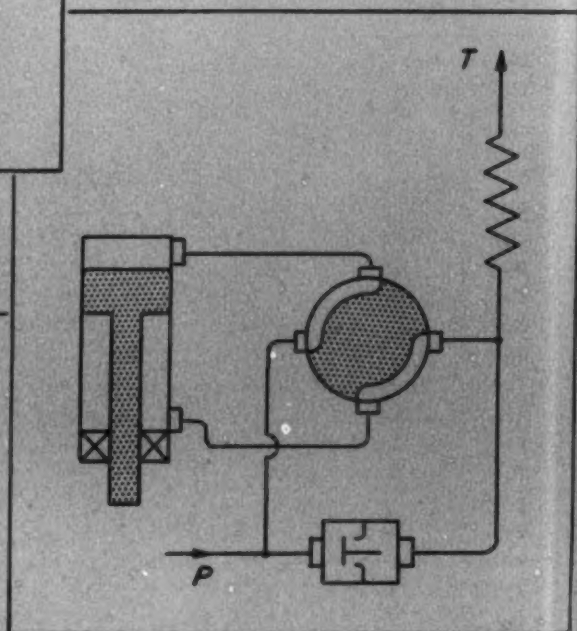


Fig. 42—Above—Alternate system for prefilling from the opposite side of the cylinder

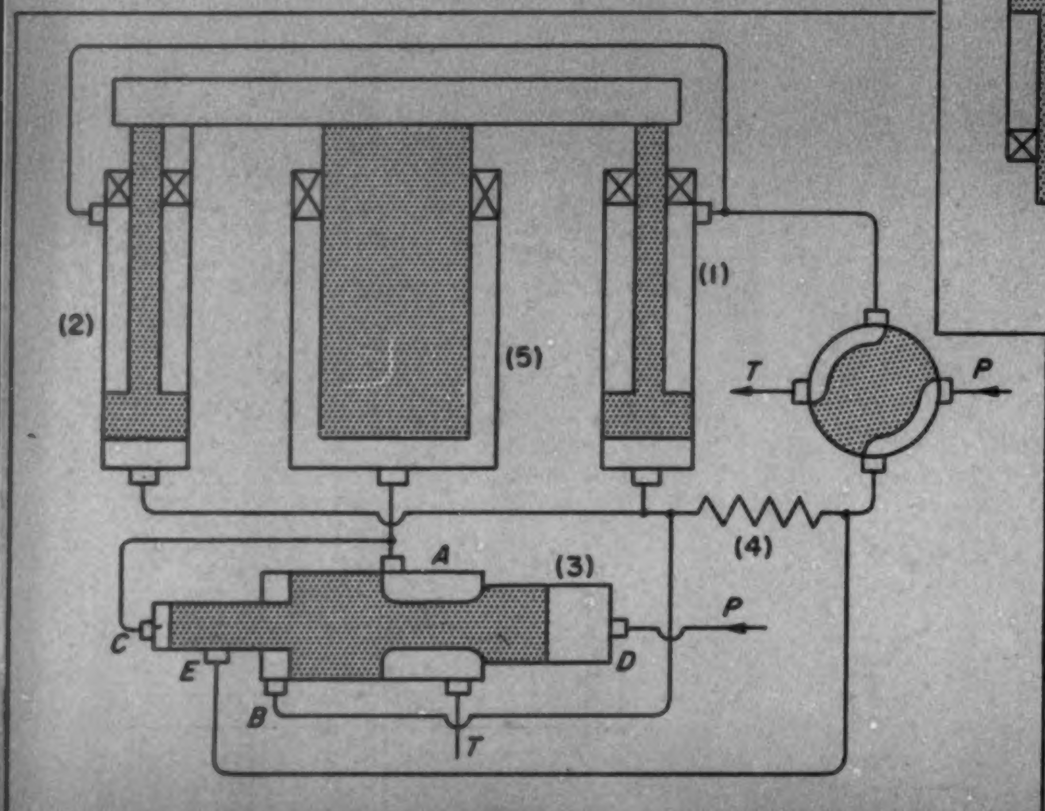


Fig. 43—Left—Prefill system designed for a press application

able periods. Load during the approach stroke is low and the loaded portion of the stroke is of negligible length. Thus, with the system adopted, the peak power is quite small and unloading in the usual sense may be dispensed with. The problem is to maintain power on the closed presses while others are being operated and to do this with the minimum waste of power. The large-output low-pressure pump *A* is used for approach and return motions and eventually blows off through the relief valve *B*. At the end of approach, pressure rises to operate the valve *C*, set to a value slightly lower than *B*. When valve *C* is operated maintaining pressure is admitted from the small pump *D*, which blows off through the relief valve *E*. A check valve *F* ahead of the selector is obviously necessary.

Variable-Area Systems: In variable-delivery systems, including twin pump, the aim is to keep the power approximately constant by increasing the output while pressure is low. Another method of keeping the power constant would be to have a constant delivery, but to keep the pressure constant by altering the effective area of the cylinder. Obviously continuous variation of effective area—the strict counterpart of a variable-delivery pump—is not feasible, at least as far as the author knows, but operation on two distinct areas is relatively easy.

The simplest method of operating on two areas is to use a cylinder "differentially" for approach, i.e., to supply pressure to both sides, thus effectively working on the area of the piston rod. A system on this principle is shown in *Fig.* 38. With the selector in the position shown, pressure is applied to the piston at the right of the approach valve (1) which causes the latter to move until ports *A* and *B* communicate (as shown), pressure then being applied to both ends of the cylinder. When pressure rises, the sequence valve (2) operates and moves to the right admitting pressure to connection *C* on valve (1), which then moves fully to the left cutting off the connection between ports *A* and *B* and connecting the latter to tank. The cylinder is now of the normal double-acting type, pressure acting on the full area of the bore. Valve (2) stays put until automatically reset when the selector is reversed.

If high speed is required during approach, the piston rod must be made small. Consequently, the area for the return stroke—the annular area between piston rod and bore—will still be quite large and return will be comparatively slow. Quick return can be obtained by using a three-chamber cylinder, *Fig.* 39, in which the end space is permanently connected to pressure. The cross-section area of the rear rod is made slightly smaller than that of the annular space at the front. During forward motion the cylinder behaves exactly in the same manner as in the previous example. During return motion, the effective area is the difference between front annular area and area of rear rod —area of bore less sum of areas of front and rear rods.

Instead of pressure control, it is also possible to use positional control to determine the changeover point between high and low speeds, exactly as in the case of two-pump systems. Rather large circulating flows are involved in systems of this type, and the pipes and control gear must be of adequate size to cope with them.

Prefill Systems: Systems of the "prefiller" type are essentially applicable to cylinders moving under negative load, usually due to external force such as gravity or to approach cylinders of small size. It is assumed that the negative load is capable of moving the cylinder faster than the pump would be able to do. Under these conditions cavitation will result and there is nothing to be gained if the vacuum thus created must be made up from the pump delivery. The vacuum must be made up either from the tank or, possibly, from the opposite side of the cylinder if the latter is double acting.

One method of prefilling from the tank is to use a check valve as in *Fig.* 40, the cylinder illustrated working under negative load during part at least of the extension stroke. The check valve must work at subatmospheric pressure and must not exert too much restriction on flow; hence, it is likely to be rather delicate and other systems are generally preferred. Positional control of prefilling either from the tank or from the opposite side of the cylinder is quite feasible and needs no comment.

Fig. 41 shows a system for prefilling from the opposite side, for a double-acting cylinder subject to negative load during extension. Under the action of external load the pressure drop in the resistance *A* is sufficient to operate valve *B* against its return spring, thsu opening a short-circuit path across the cylinder. The amount of fluid returned to the opposite side under these conditions can be increased by adding another resistance *C*. Both *A* and *C* could also be of the one-way restrictor (restriction valve) type to avoid obstruction during retraction. Complete prefilling is obviously impossible except under conditions where the negative load acts during *retraction*. When the negative load vanishes and the cylinder slows down, the pressure loss across *A* (now governed by the output of the pump) is small enough to allow the valve to return. In practice, the resistance *A* would be an orifice in the operating piston of the valve.

Another system for prefilling (at least partially) from the opposite side of the cylinder is shown in *Fig.* 42, which is self-explanatory. This system works in either direction, but is rather inefficient, particularly if the selector is far away from the cylinder.

Fig. 43 shows a system for prefilling from the tank, applicable to a press. Quick approach is obtained by two small auxiliary cylinders (1) and (2). When the latter are extending, pressure at port *B* of the prefiller valve (3) is smaller than at port *D* due to the drop across the resistance (4), and the valve remains in the left-hand position with port *E* being blocked and port *A* connected to tank, thus allowing the main cylinder (5) to fill from the tank. When load is encountered, flow through resistance (4) stops, pressure at *B* rises. Proportions of the valve are such that, under these conditions, it will move over to the right to closing port *A* and connect port *E* to port *C*, thus putting the main cylinder to pressure. The valve remains to

the left during the return stroke which is effected by the auxiliary cylinders, the main one being single-acting.

Auxiliary approach cylinders are often arranged to be inside the main cylinder, as in *Fig.* 44; in this case the annular area of the main cylinder is used for return. Constructions of this type are usually found on presses.

Accumulator and Intensifier Systems: Accumulators can be used for quick approach under low load, with a high degree of power economy. A high-pressure pump may be used to store low-pressure potential energy by using the special form of accumulator shown in *Fig.* 45. In this design the low-pressure chamber is filled from the tank during the storing stroke. The latter must still take place during an idle period but the time required will be much shorter than it would be if the same volume had to be stored at full pressure.

Intensifiers can also be used to advantage as a solution to the approach problem, at least in cases where the work to be done under high load is relatively small. The pump must then be designed to suit the approach portion of the stroke, being in fact equivalent to the larger pump of a two-pump system. A simple system of this type is shown in *Fig.* 46; the intensifier *A* is automatically reset before or at the end of the approach period. When resistance is encountered by the cylinder, pressure rises until the sequence valve *B* operates, bringing the intensifier into action. The check valve *C* is obviously essential for the system to function at all. The relief valve *D* is set at a pressure lower than *B* and prevents the latter from operating during the return stroke.

Intensifiers are of course particularly useful for machines designed to work at high pressures, enabling these to be reached without using high-pressure pumps, which tend to be rather expensive.

It is also possible to use a high-pressure pump only, with an inverted intensifier for approach. An example is shown in *Fig.* 47. The inverted intensifier *A* has a volume on the low-pressure side slightly in excess of the amount required for the approach portion of the cylinder travel and is returned by a spring during or after the return stroke. It is refilled from the tank through a check valve *B*. When load increases, full

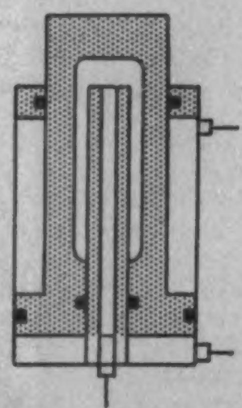


Fig. 44—Above—Auxiliary approach cylinders arranged inside the main cylinder for presses

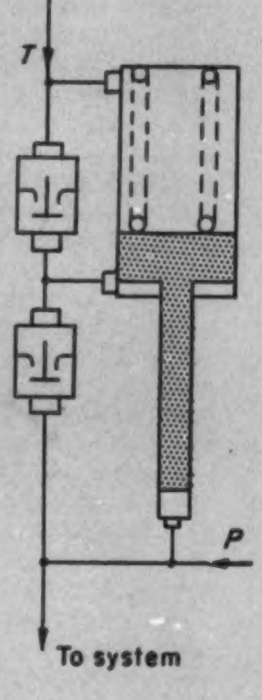


Fig. 45—Center—Special form of accumulator using a high-pressure pump to store low-pressure energy

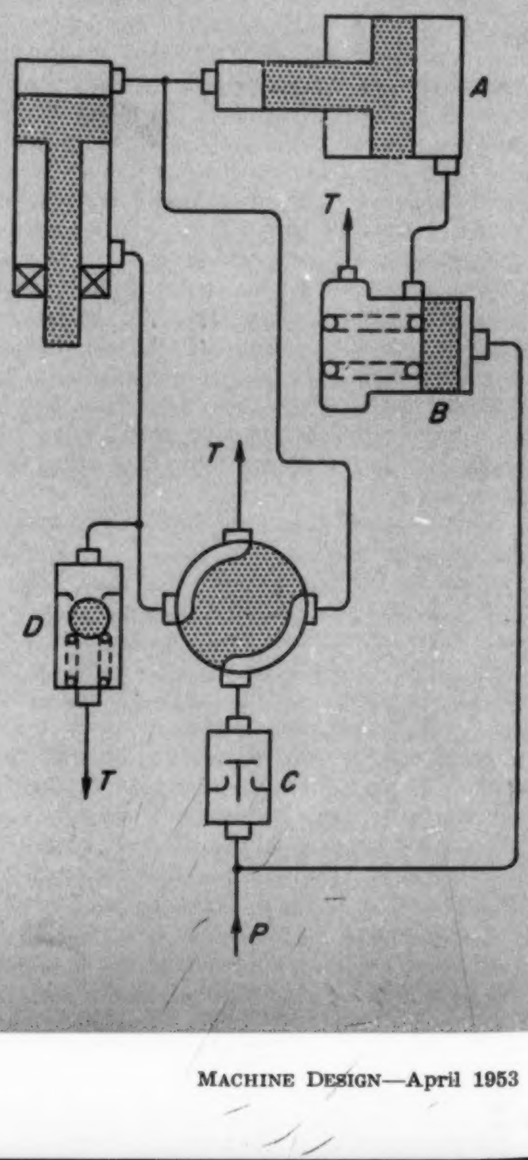


Fig. 46—Right—System employing an intensifier as a solution to the fast-approach problem

pressure reaches the cylinder through the sequence valve *C*. This system would be rather cumbersome unless the cylinder volume is quite small. It might not be a bad solution, say, for power brakes and a few other similar cases.

Geometric Efficiency and Load Balancing: In a few special instances the load on a cylinder may be varied within limits. Perhaps the only major instance of this arises in the case of aircraft undercarriage retraction. In *Fig.* 34*b* is given a typical load-stroke curve and defined geometric efficiency. The work to be done, shown by the area of the curve, may be fixed but the shape of the curve is a function of the linkage interposed between the cylinder and the load to be lifted or, more broadly, of the geometry of the whole system. The usual aim is to reduce the peak power required or, for a given power, to increase the speed of operation by increasing the geometric efficiency of the system, i.e., by reducing the ratio of peak to average loads. Undercarriage retraction mechanisms are often the subject of several redesigns to improve their characteristics in this respect.

In this kind of application it is also often feasible to reduce peak loads by partial balancing. For aircraft purposes, balancing weights are of course out of the question, but air-loaded balancing cylinders or other elastic devices may be used. These are arranged to reduce the *peak* load as much as possible, even at the cost of increasing the total work done during the

Energy Storage and Central Systems: In many hydraulically operated machines or groups of such machines, power is required only at intervals for comparatively short spaces of time and with relatively long idle periods in between. If the pump is unloaded during idle periods, there is no *energy* to be economized by power storage but a saving can still be made in the *peak power rating* of the driving means, and usually also in that of the pump by making use of idle periods to accumulate a store of energy.

Storage of energy in a flywheel in the pump drive is feasible as a means of reducing the peak power rating of the driving motor (not that of the pump), if peak power is required for very short periods only and if the pump is unloaded during idling periods. In practice, the system is hardly applicable if the pump is to operate a number of independent services or if there is any danger of the intended full power period being exceeded, since this would cause stalling of the driving motor. Flywheel storage is, however, quite feasible in a few special cases.

The usual method of storing power is in a hydraulic accumulator, which is almost invariably (but not necessarily) used in conjunction with an automatic cutout. Accumulators of low capacity may be used to maintain pressure during idling periods, but these would not be described as "energy storage means" in the sense intended here, which requires the accumulator to be large enough to perform complete operations. For this purpose, the accumulator must have a fluid capacity equal to the volume of the largest cylinder in the system or of several together, if two

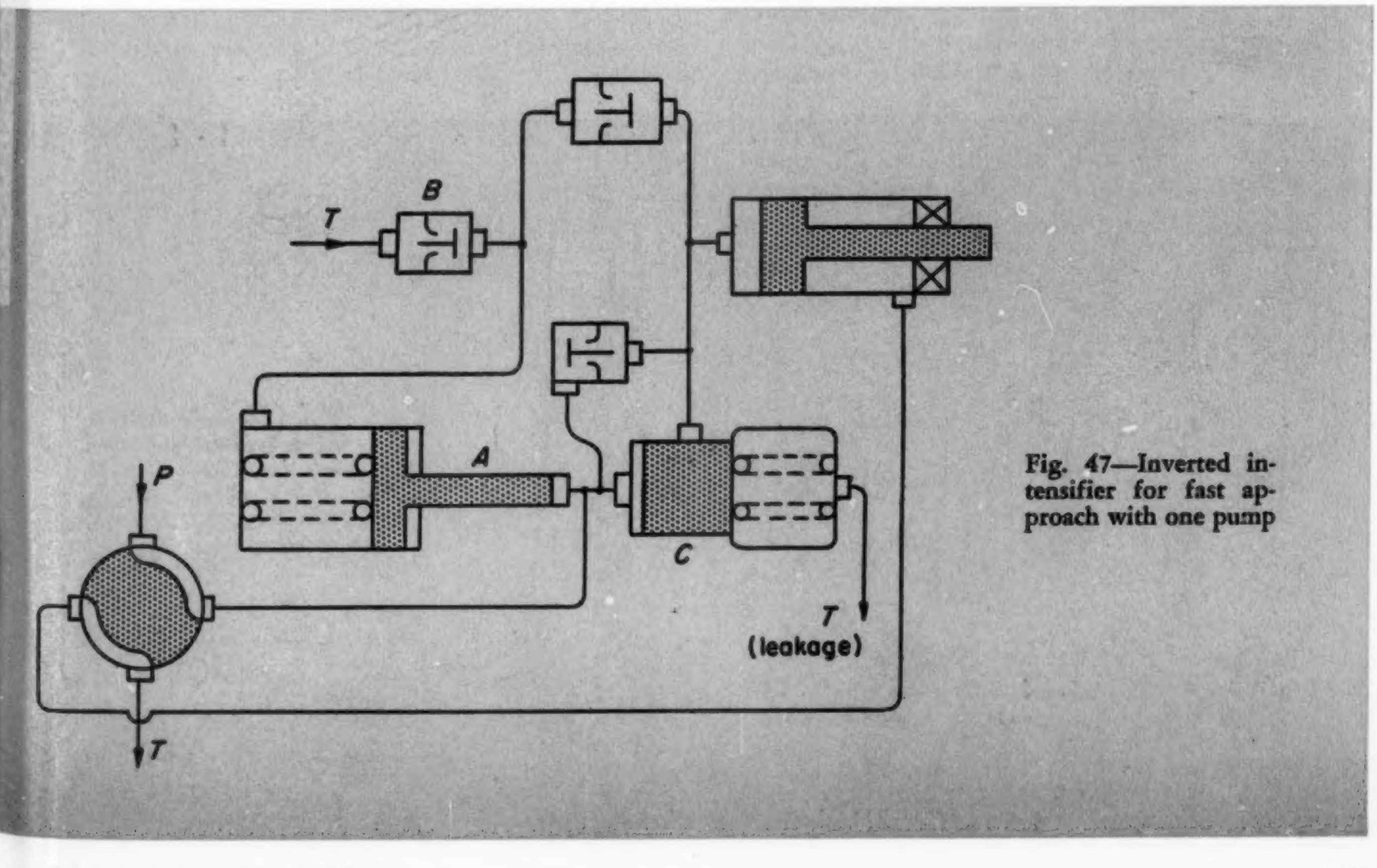


Fig. 47—Inverted intensifier for fast approach with one pump

or more are to be operated simultaneously or in close succession. The size of the pump is then determined by the time available for recharging, i.e., the minimum idling period or, in the case of a number of independent services, by the average consumption with a suitable margin to ensure that the accumulator will never empty under normal conditions.

Accumulators are sometimes used in this manner to obtain rapid operation without using an excessively large pump, as in die casting machines. They are also not uncommon in aircraft hydraulic systems.

Another example is found in press shops where hydraulic mains are often used to operate a large number of presses, working independently of each other. In such instances "live" mains and an ample store of energy are essential. This type of system is tending to disappear, however, in favor of presses with individual pumps.

CONTROL OF LOAD SPEED AND POSITION

Fundamentals of Control: The basic hydraulic control operation is that involved in starting or stopping a motor element and determining its direction of motion. In the case of a continuous hydraulic motor, if motion is always in the same sense the only operations involved are starting and stopping and these can be carried out by a stop valve, *Fig. 48a*. When the valve is closed the motor is hydraulically locked against backward rotation. If the motor is to be freed, a three-way selector may be used, as at *b*, having three positions corresponding to "start", "stop", and "free". For a reversing motor a four-way selector should be used, *Fig. 48c*, and according to the type of neutral provided, the motor will be free or locked in the central position of the selector lever.

A single-acting cylinder is usually controlled by a

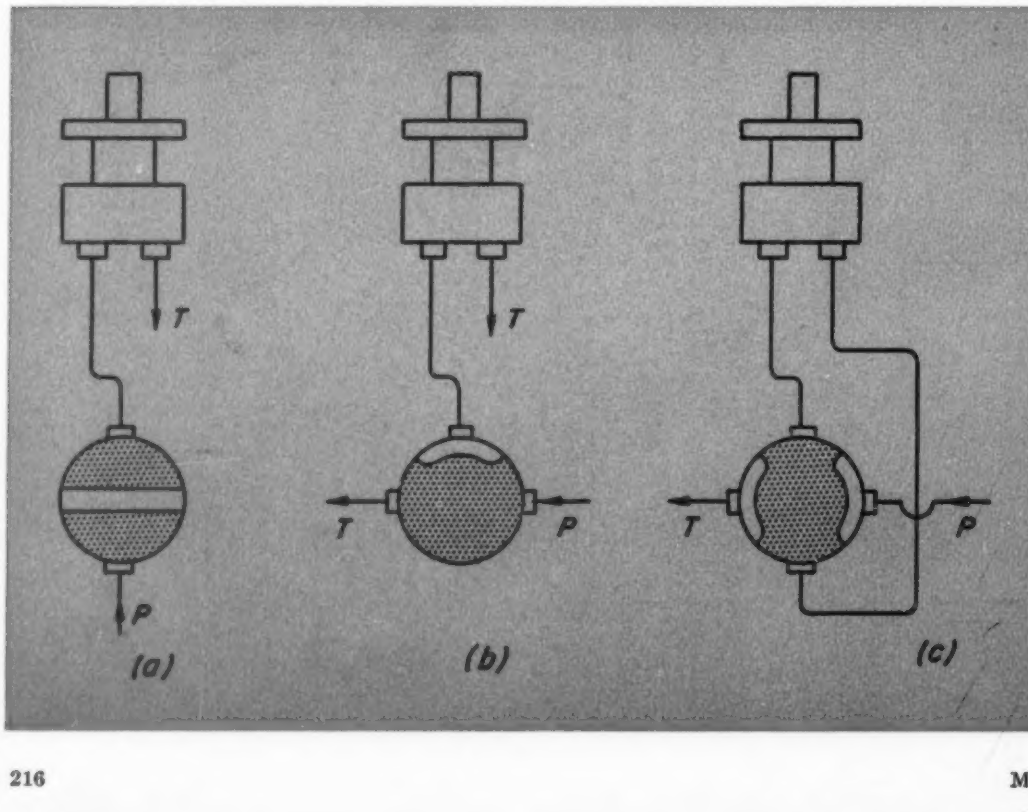


Fig. 48—Basic control of a hydraulic motor for start and stop is shown at *a*, with a "free" position at *b*, and for reversing at *c*

three-way selector, *Fig.* 49. If the latter has three positions, the cylinder will stop in "neutral" unless it is subject to negative (tension) load. Single-acting cylinders are often gravity returned, as in hydraulic lifting gear and upstroke presses, although in the latter case the available return pressure (that corresponding to the weight of the platen) may be rather low. Spring return is also feasible in many cases if the work to be done (usually against mechanical and hydraulic friction, sometimes also against external loads) is not too large. Size of steel springs might be rather cumbersome if the work they must do is at all appreciable. Also, other forms of spring are used sometimes—rubber cords to effect extension of retractable aircraft undercarriages.

The lightest and most compact spring medium is undoubtedly compressed air or other gas, and this is also used to effect return of single-acting cylinders. Probably the simplest arrangement is that in which one side of the cylinder (which is of double-acting construction) is connected to an air accumulator, a volume of oil being interposed between the air and the cylinder to lubricate the seals and prevent air leakage into the opposite side, *Fig.* 50. In this kind of application a separator piston or diaphragm in the accumulator is unnecessary, as the air is always in contact with the same volume of oil, and does not tend to be dissolved away.

Return may also be effected by a single-acting cylinder of smaller area permanently connected to pressure. A more frequent method of achieving the same result is shown in *Fig.* 51, this system being generally referred to as "differential"; when the cylinder extends, pressure effectively acts on the area of the piston rod.

Fully double-acting cylinders are generally controlled by four-way selectors, *Fig.* 52. With a "blocked" type neutral as shown, the cylinder is hydraulically locked in the neutral position. If in the neutral position of the selector one or both lines are to tank, the cylinder is free to move in one or both directions under the action of external load. A neutral position with one or both lines to tank is sometimes used, not with the object of leaving the cylinder free to move but to prevent pressure build-up due to thermal expansion or leakage in locked lines and sometimes also, as in the case of open-center systems, because it may simplify selector design. The fully free neutral—both lines to tank—is also occasionally used to leave the cylinder free to float in some special applications such as earth scrapers. In such cases it may be necessary to provide both "locked" and "free" neutrals, involving a four-position selector, in the circuit.

Resistance Control: *Fig.* 53*a* shows a single-acting cylinder permanently connected to tank through a resistance and controlled by a two-way valve. When the latter is closed the cylinder is free to return (under external load) at a limited speed. When the valve is open the resistance maintains sufficient pressure to operate the cylinder, although some of the pump output is of course wasted. Shown at *b* is a system on the same principle, but here the resistance connects the cylinder to pressure and the blocking valve connects it to tank, extension being effected when the valve is *closed*.

Resistance control is quite generally applicable in theory and calls for one less connection on the control valve than would otherwise be necessary. This factor is more important with seating than with port valves, since the complexity of the former increases rapidly with the number of connections.

Resistance control may be used on a double-acting cylinder, *Fig.* 54*a*, which extends when the blocking valve is closed and retracts when it is open (the valve could of course be interchanged with any of the three resistances). When the valve is partially open, the circuit is that of a Wheatstone bridge and balance is achieved at only one particular degree of opening. Very sensitive control can be obtained by this means if the cylinder and load have low friction. This type

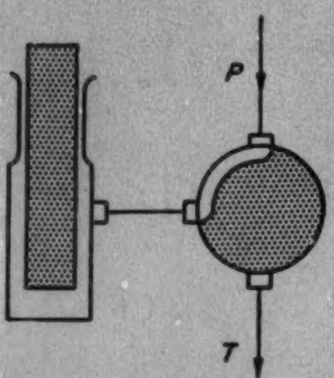
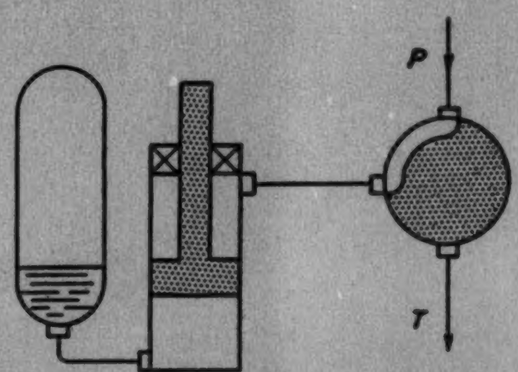


Fig. 49—Left—Single-acting cylinder controlled by one three-way valve

Fig. 50—Right—System for accumulator return of single-acting cylinder arrangement



of control is sometimes used in presizing or copying attachments for machine tools, in conjunction with a follow-up servomechanism[5, 6, 7].

In theory at least, still greater sensitivity can be obtained by the use of the type of selector shown in *Fig.* 54*b* (or its seating valve equivalent). In the neutral position there are very small passages from each cylinder line to the pressure and tank connections. The circuit is still that of a Wheatstone bridge but all four resistances are varied at once. In practice, at least for port type selectors, there is some difficulty in making the passages small enough in the neutral position and fine tolerances may be involved. With a seating valve selector, the four openings may be adjusted individually, but operating loads may be too high for some applications. For these reasons, a two-way throttle valve, as in *Fig.* 54*a*, may often be the most convenient solution if extreme sensitivity is the aim and if the power involved is low.

Resistance control is, of course, wasteful of power and for this reason is most commonly used for controlling large valves by means of smaller pilot valves. Where there are multiple control elements, the circuit may differ appreciably if control is of the ordinary or of the resistance type. For instance, in the fairly common occurrence on presses, where an unloading valve is to be controlled by a number of pilot valves, when any one is operated the main valve closes, closure being effected by putting the control connection of the main valve to pressure. With ordinary control, the pilot valves must be in series as in *Fig.* 55*a* but with resistance control they must be in parallel as at *b*.

The use of resistance control, or more generally of "hydraulic potentiometer" arrangements, is of wide applicability. *Fig.* 56*a* shows a method of obtaining a small flow at a reduced pressure p_1, the system pressure being p_0. The resistances R_1 and R_2 may both be fixed if p_1/p_0 is fixed or one or both resistances may be variable. In the latter case p_1 may be applied to a spring-loaded member, usually the operating piston of a valve, to cause the latter to move by an amount roughly proportional to the ratio $R_2/(R_1 \times R_2)$ if p_0 is fixed, *Fig.* 56*b*.

Simultaneous Operations: In the most general type of hydraulic system there are usually a number of cylinders, or other motor elements, with independent or interdependent control elements. Consider for instance the circuit shown in *Fig.* 57 having two cylinders, *A* and *B*, controlled by two independent selectors, *a* and *b*. Cylinder *A* is shown in the diagram as being in motion, selector *b* being in neutral, and/or cylinder *B* being on its stops (fully extended or retracted).

Assuming that the source of power is a displacement pump—a source of fluid volume and not of pressure—let selector *b* be operated to move cylinder *B*. If the pressure required to operate *B* is greater than that required to operate *A*, the former will remain stationary while the latter completes its travel; *B* will then move. If the pressure required to operate *B* is smaller, in the circuit shown *A* will actually move backwards (unless the load on it is pure friction or some equivalent) while *B* moves forward. This may be prevented by fitting check valves at the pressure connection of the selectors, as shown in *Fig.* 58. With the latter circuit, assuming again that the load on *B* is relatively smaller, *A* would stop when selector *b* is operated, and *B* would have to complete its stroke before cylinder *A* moves again.

The general principle may be expressed thus: In a system where two or more paths are open to the output of a displacement pump, the pressure in the delivery line will drop to the value corresponding to the path of least resistance. In the example, the effect of resistance of the pipes and valves has been neglected because it is usually too low (unless deliberately

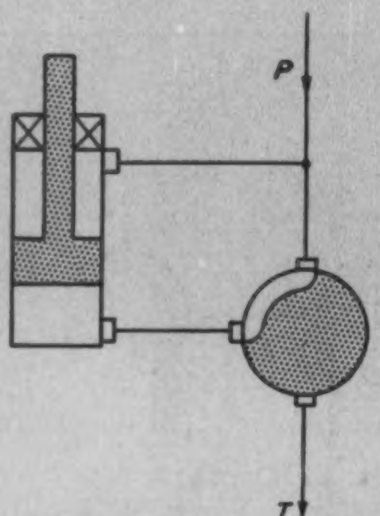
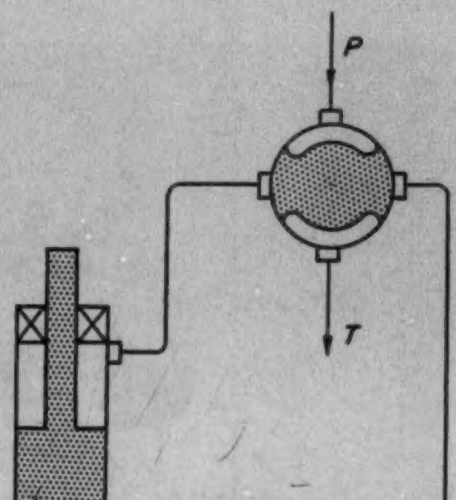


Fig. 51—Left—Circuit for accomplishing the same single-acting result as that of *Fig.* 50 using permanent connection to pressure

Fig. 52 — Right — Typical four-way valve controlled double-acting cylinder



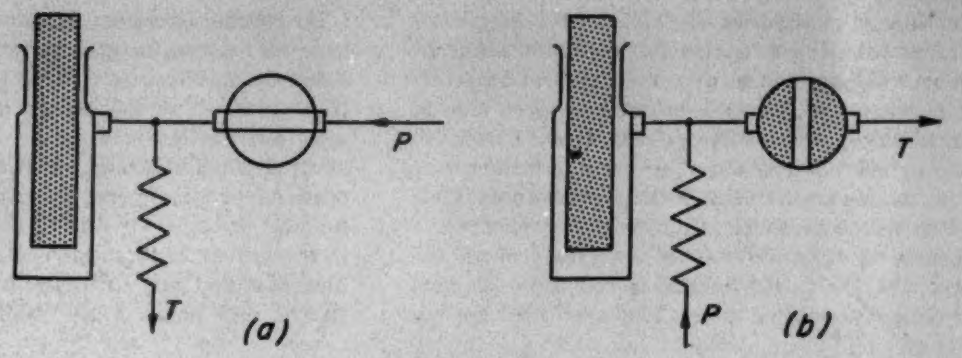


Fig. 53 — Connection of cylinder permanently to tank through a resistance, as at *a*, controls free return and, at *b*, resistance connects the cylinder to pressure

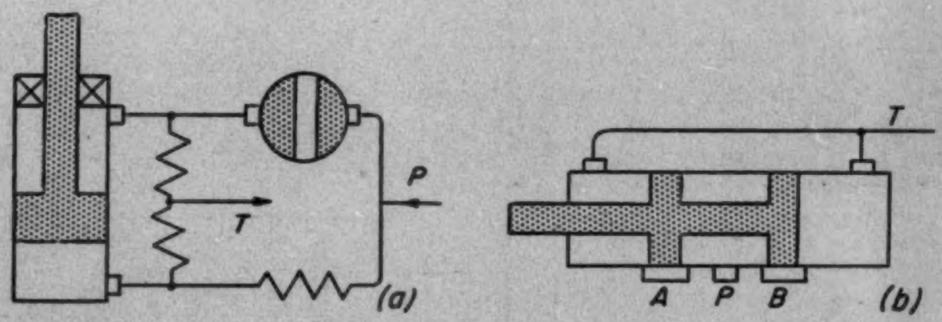


Fig. 54 — Resistance control of double-acting cylinder, *a*, to give a Wheatstone bridge circuit at partial opening. Greater sensitivity is theoretically possible with the selector shown at *b* having restricted passages in neutral

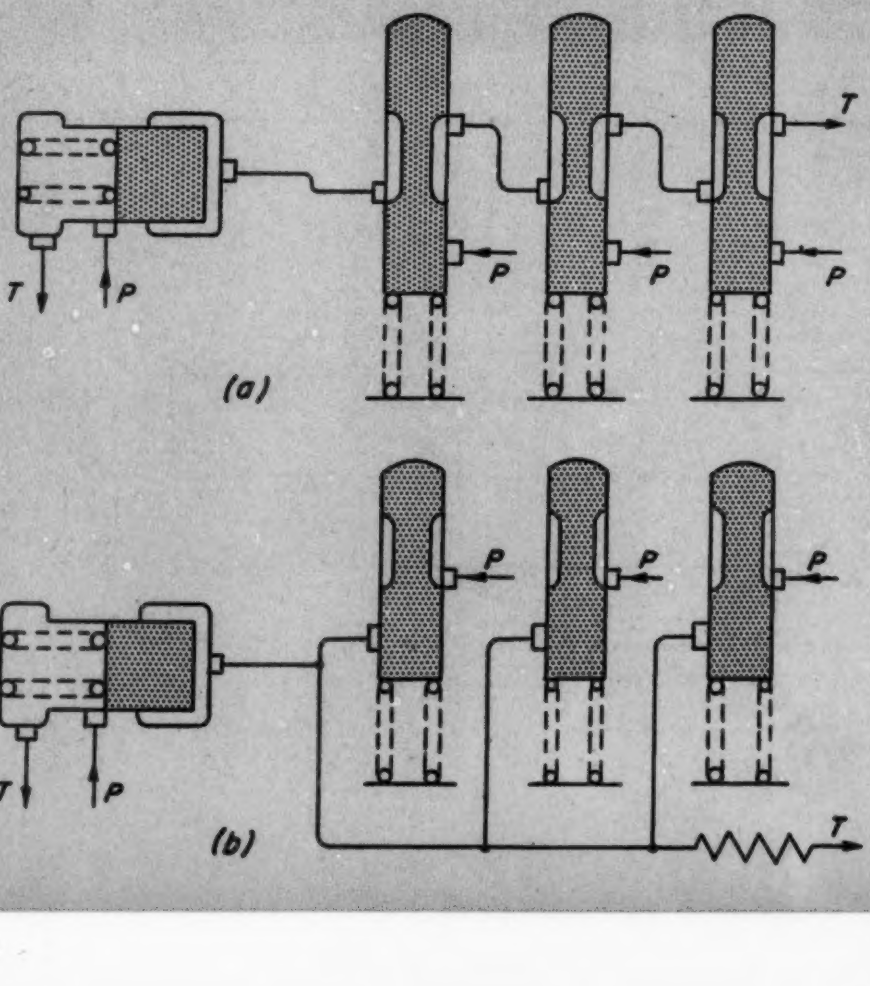


Fig. 55 — Press circuit with multiple-pilot control of unloading as commonly arranged at *a* and with resistance control at *b*

increased) to have any appreciable effect.

The problem is exactly the same if there are a number of cylinders in parallel controlled by a single selector. However, the situation is somewhat different if the source of power is an accumulator or a centrifugal pump, when conditions approximate more closely those obtaining in an electric circuit where (pressure being the primary datum) flow to various branches is shared in inverse proportion to their resistances. This is also the case if pressure is limited by a relief valve (as it nearly always is) and the resistance of all the paths open to the pump output is too large to take the full delivery, part of which blows off through the relief valve. Solutions will be detailed later.

Hydraulic Locking: If the selector has the right type of neutral, a cylinder may be locked in any position by moving the control lever to the neutral position (usually at the center of its travel). In many selectors the lever is spring centered and returns to neutral when released. However, where the cylinder need never be stopped in midtravel, the necessity of holding the selector during the motion or of returning it to neutral, if it is not spring centered, is often undesirable and selectors may be of the "stay put" type. In the end position the cylinder is still held on its

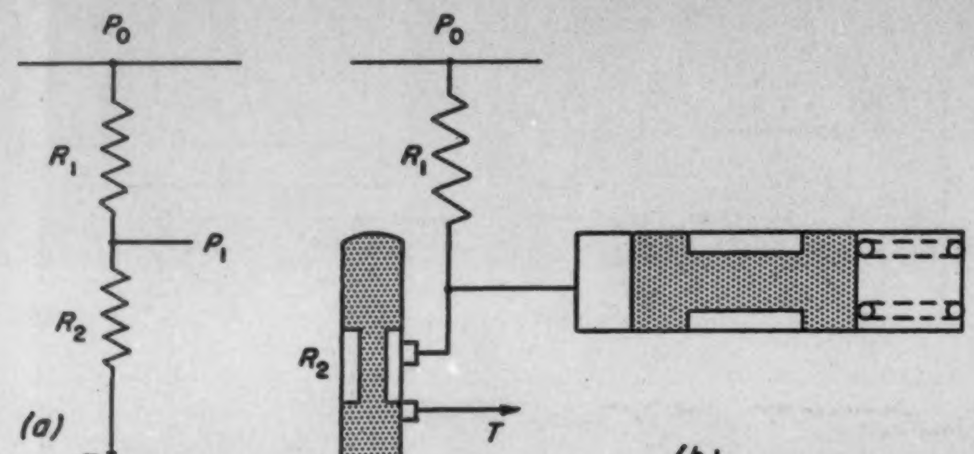


Fig. 56—Typical "hydraulic potentiometer" circuits

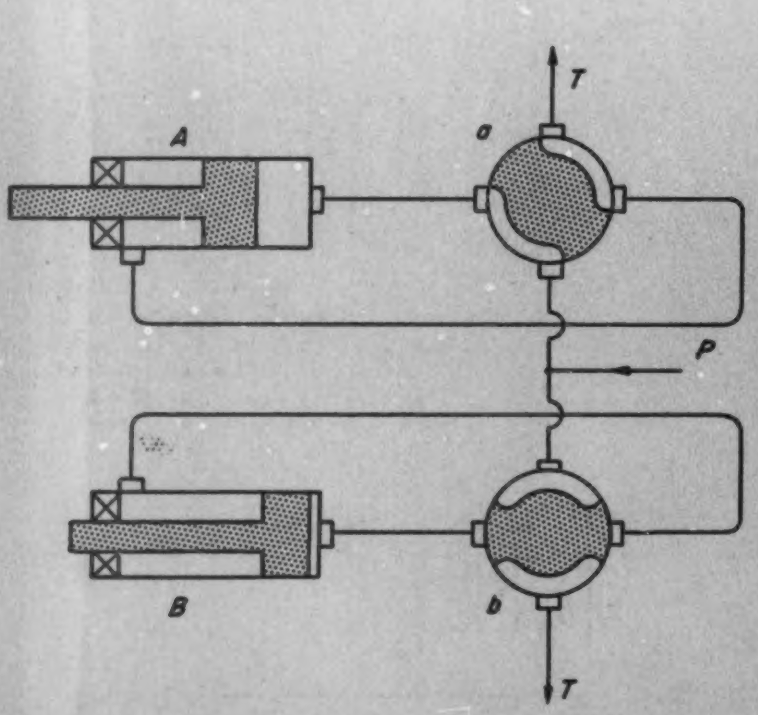


Fig. 57—Multiple-cylinder system with two independent control valves

stops by hydraulic pressure, if available, but the following cases must be considered:

1. When stationary, the cylinder may be called upon to resist higher loads than those against which it has to move. Such loads could move the cylinder by blowing off oil through the relief valve in the system.
2. Pressure may temporarily drop due to operation of another cylinder.
3. Pressure may drop when the pump is unloaded, if the circuit is of the dead type.

If any of these possibilities may occur, the cylinder can still be locked in its end position, without neutralizing the selector, by means of a check valve at the pressure inlet to the latter, as shown in *Fig.* 58. Note also that "differential" cylinders may be locked at both ends in the same manner, *Fig.* 59. With the selector and cylinder in the position shown, the oil is trapped by the selector and the check valve. In the opposite position, locking is effected by the check valve only, since the cylinder could not move without returning fluid to the pressure main.

The effectiveness of the types of locking discussed so far depends on the freedom from leakage of the selector. In many applications a limited degree of "creep" of the cylinder is permissible and in such

Fig. 58—To prevent reverse movement of one cylinder in the circuit of *Fig. 57*, a check circuit at the pressure connections of the valves can be used

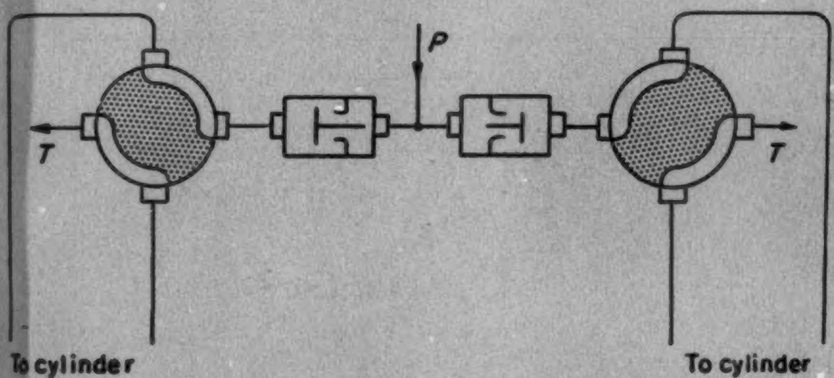
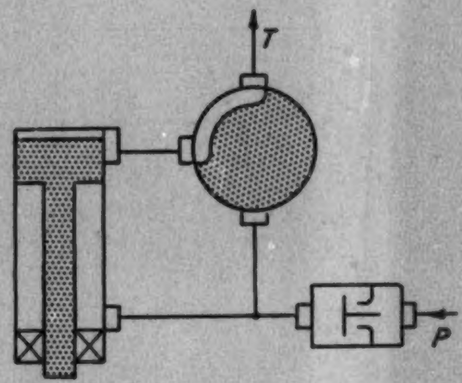


Fig. 59 — Method of locking differential cylinders at both ends



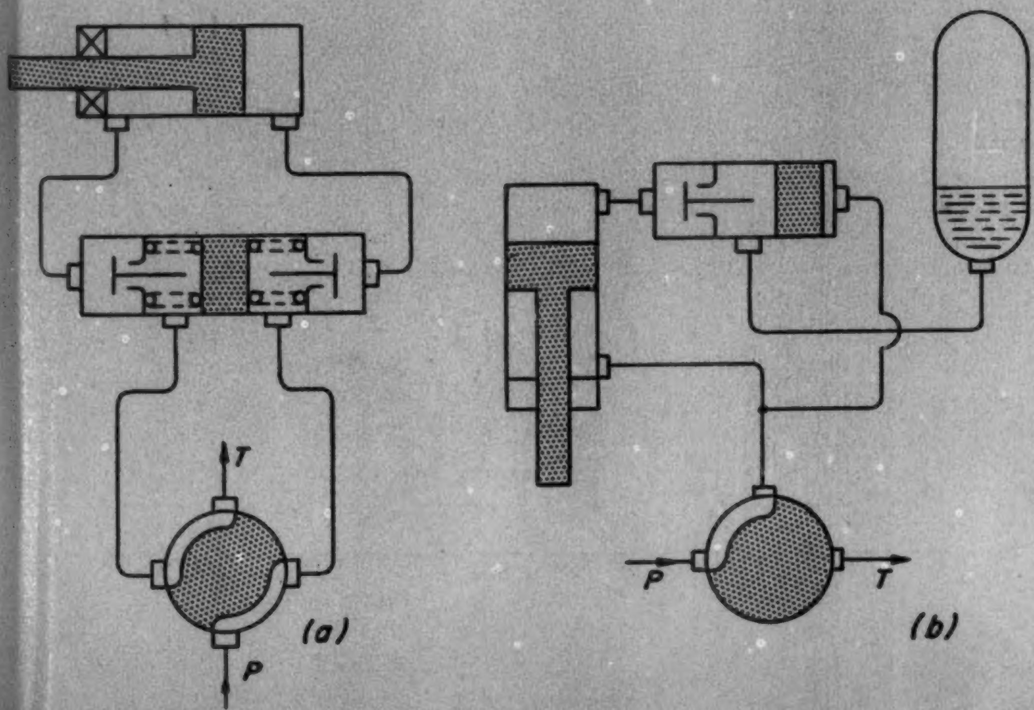


Fig. 60 — A circuit with a double hydraulic locking valve is shown at *a* and one for an accumulator return circuit at *b*

cases a port type selector, although it usually allows some leakage, is often effective enough. Where the leakage through the selector is excessive and in certain cases in aircraft where it may be desirable to guard against breakage of pipes, use may be made of hydraulic locking valves which are almost invariably of the seating type and, therefore, allow little leakage, if any. *Fig.* 60*a* shows a circuit having a double hydraulic locking valve. The valve on the return side is opened by pressure in the opposite line, while the other valve is lifted by the flow. Hydraulic locking valves may also be used in circuits having accumulator return, as in *Fig.* 60*b*, in which locking in the "return" position cannot be effected by the selector.

The hydraulic locking valves illustrated do not represent a new type of valve, but are identical with

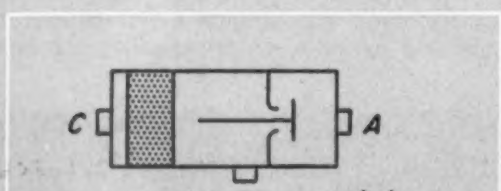


Fig. 61—Left—Pilot-operated two-way valve, *a*, of identical basic design as the locking valves of *Fig.* 60 and the preferred design, *b*, to obviate pressure upset

Fig. 62 — Below — Relief valve application to avoid thermal pressure overload of the circuit of *Fig.* 60*a* is shown at *a* and one for a selector-locked circuit at *b*

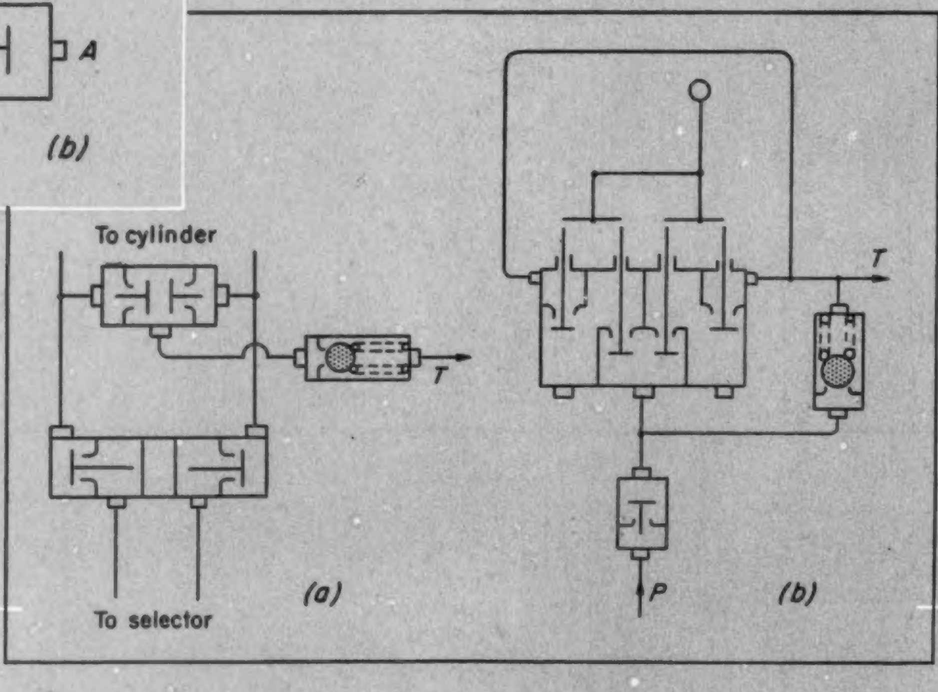
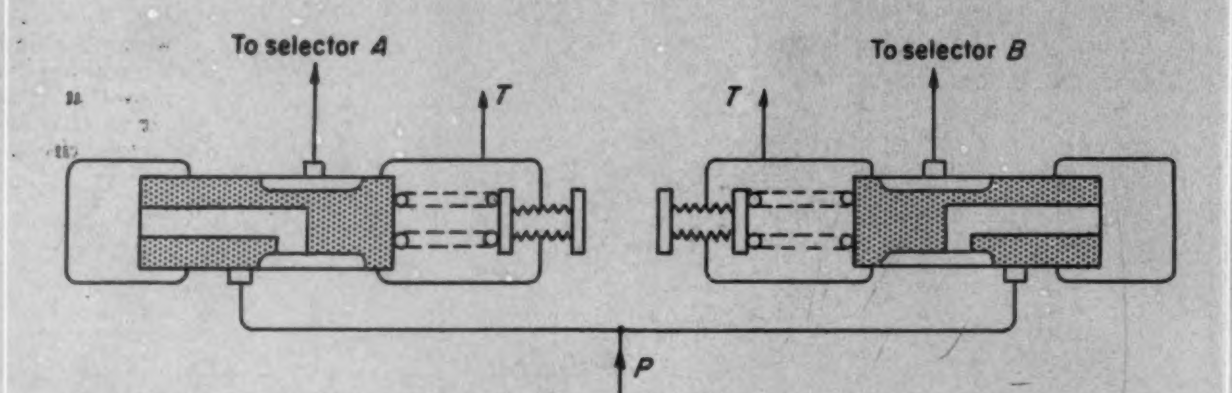


Fig. 63 — Below — Circuit for a machine with reducing valve adjustment for pressure regulation to each cylinder



the pilot-operated two-way valve of *Fig. 61a*. The valve of *Fig. 61b* is often used in preference, as it is not likely to be upset by pressure in the return line.

In common with other locking systems such as irreversible worm drives, a hydraulic locking arrangement such as that of *Fig. 60a* tends to exhibit dither when the cylinder is moved against negative loads. The analogy with mechanical devices, some of which have succeeded in overcoming this phenomenon, suggests that the trouble is not fundamental but could perhaps be cured by a sufficiently gradual opening of the valve and/or some degree of damping.

Hydraulic locking operates by blocking a certain volume of fluid, which thus becomes isolated. If temperature rises, this body of fluid may be subject to severe pressure rise due to thermal expansion. Expansion of the pipes, cylinders, etc., does little to compensate. If locking is effected by port type valves, the latter usually allow sufficient leakage to prevent appreciable pressure being generated. If seating valves are used, however, thermal effects may be dangerous unless provision is made for relief. Such provision is usually in the form of relief valves, which need only be of small size. *Fig. 62a* illustrates the application of a relief valve to the circuit of *Fig. 60a*. *Fig. 62b* illustrates the application to a circuit where locking is effected at the selector. A neat construction results if the thermal relief is built in the locking valve or selector.

Control of Load: In almost any power-driven hydraulic system, there is a problem of pressure limitation in the sense of preventing excessive pressure rises when the resistance encountered by a cylinder or other motor element becomes indefinitely large. Relief or cut-out valves and various other devices are used as the pressure-limiting means. In handling machinery, lifting gear, most machine tools, punching and blanking presses, and various other applications the external load is the datum and the relief valve or cut-out must be set to a pressure somewhat higher than that required to overcome the load. Full pressure is exerted only when the cylinder, or the mechanism actuated by it, reaches its stops.

In many machines of the press or vise types, in brakes and in general wherever the work forms the stop, the converse is true. The load exerted on the work is determined only by the setting of the pressure

Fig. 64—Below—Example of a circuit designed to compensate for decompression effects

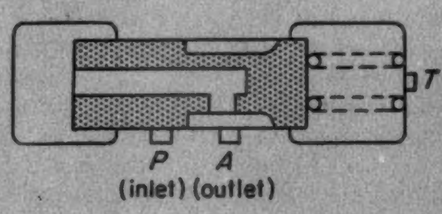
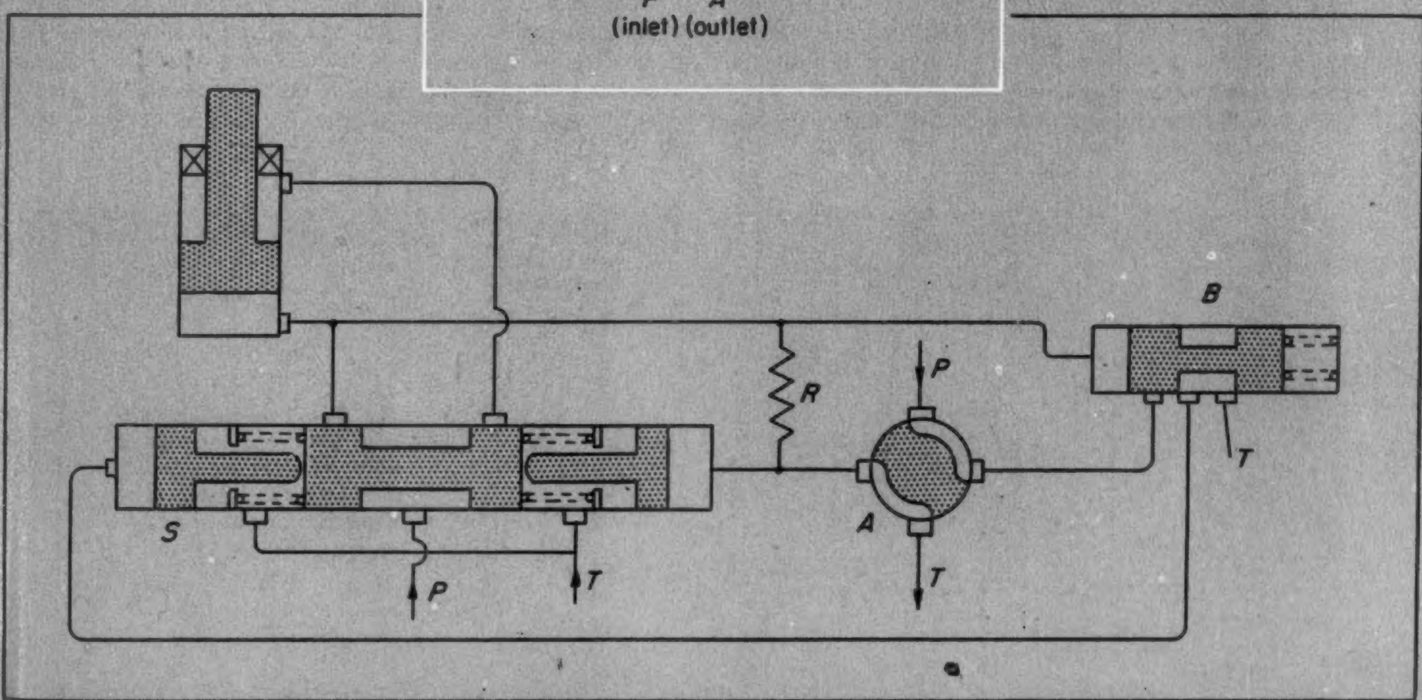


Fig. 65 — Left — Reducing valve for an accumulator type of hydraulic brake system



regulating device, which is often made adjustable.

Thus in the simplest case of machines of the second type, load can be regulated by adjusting a relief valve. If the immediate source of power is an accumulator loaded to a given pressure, regulation by relief valve is no longer feasible and it is necessary to use a reducing valve. More generally, reducing valves must be used when it is required to regulate pressure in one or more cylinders to some value *lower* than that which may obtain in other parts of the system. *Fig.* 63 shows a circuit for a machine in which pressure on the two cylinders must be adjustable independently, and either setting may be the lower of the two. If the pressure on one of the cylinders were to be always higher than on the other, the corresponding reducing valve could be omitted and the main system relief valve made adjustable.

The converse problem arises when it is required to maintain a pressure not less than a specified minimum value in part of the circuit despite the existence of lower pressures elsewhere. This problem also arises in connection with pilot control systems and a solution can be found in the use of pressure-sequence valves. Solutions for other cases follow the same lines.

In large presses, release of load may be something of a problem since the elastic energy stored in the oil and possibly in the machine frame may be considerable; its sudden release may cause shocks and other undesirable effects. Various "decompression" systems have been proposed to deal with this problem, and one example is shown in *Fig.* 64. Selector *S* is spring centered and controlled by the pilot valve *A* and the system is shown in the position reached just after reversal of the pilot valve. As long as there is pressure at the bottom of the cylinder, the sequence valve *B* remains operated and both sides of the selector are to tank; the selector goes to neutral and remains there until pressure has leaked out of the cylinder through the resistance *R;* valve *B* then resets, and the selector moves fully to the right.

A simple case of machinery based on load regulation is that of power brakes. In the case of hydraulic operation, a continuously circulating hydraulic pump could be used in conjunction with control by adjustable relief valve, if reliance on the pump were acceptable. Usually an accumulator is employed and actuation of the control lever or pedal has the effect of varying the load of a regulating spring in a reducing valve which must be of the type of *Fig.* 65 (its seating valve equivalent or any other equivalent type) to allow the brakes to release. In some systems, the brakes are held off by pressure and applied by springs with the object of giving automatic braking if pressure fails; brakes would then be applied by *decreasing* pressure, but otherwise the circuit principle remains the same.

Servo controls of the follow-up type, usually vacuum operated, are also in extensive use for vehicle brakes in spite of their relative complication as they allow a direct mechanical connection to be retained in the event of power failure.

Speed Regulation with Fixed-Delivery Pumps: The vast majority of hydraulic installations are based on fixed-delivery pumps working at constant or nearly constant speed and therefore giving a constant flow. If it is desired to reduce the speed of a cylinder or motor from the value corresponding to the full pump output, this can be done at the cost of wasting some power by by-passing part of that output back to tank, or into the accumulator if one is present. For the sake of simplifying descriptions, it will be assumed that there is no accumulator in the system and that pressure limitation is effected by a relief valve.

There are two basic methods of achieving the desired object. The first method is to throttle the flow to a degree sufficient to reach the maximum pressure, when part of the pump output will blow off through the relief valve, *Fig.* 66*a*; the second is to open a bypass path through a resistance *Fig.* 66*b*. The first method requires the pump to work at maximum pressure, and is therefore wasteful of power if the cylin-

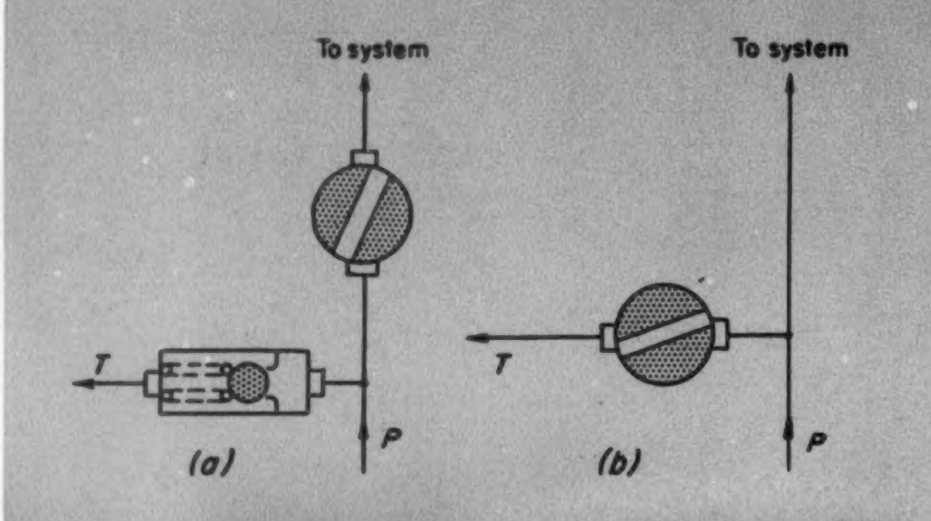


Fig. 66—Two basic methods for reducing speed of a cylinder in a fixed-delivery system by throttling, as at *a*, or bypassing through a resistance, *b*

der load is comparatively low. The second method of course also wastes power, but to a much lesser extent if pressure is far below the maximum.

With by-pass metering, the efficiency of the system is the ratio of actual to maximum *speed*. With throttle metering in the absence of an accumulator, i.e., when the excess flow is blown off through a relief valve, the efficiency is the ratio of actual to maximum *power*. With throttle metering, when the excess flow passes into an accumulator, the efficiency is the ratio of actual to maximum *pressure*, if the energy stored in the accumulator is counted on the credit side.

In either case, the setting of the resistance for a given flow will depend on the cylinder load. For many purposes it is preferable to have a device which can be set to give a predetermined speed irrespective of cylinder load—a metering valve. The general principle of most metering valves is the same; the flow to be metered passes through a resistance and the resulting pressure differential is made to act on a spring-loaded piston, which in the simplest case (as illustrated in the diagrams to follow) is the spool of a slide valve that causes the valve to open or shut to the exact extent required to balance the spring load.

A by-pass type of metering valve is shown in *Fig. 67a*. The type shown in *Fig. 67b* may be used as a throttle valve by placing it in the pump delivery line or any other required place in the line of flow. It may also be used as a by-pass type by connecting it in a branch leading to tank. Both types may be adjusted, if adjustment is called for, either by regulating the resistance (probably the commonest and best method) or the spring load. In the latter case, or if no adjustment is required, the resistance may take the form of an orifice in the piston. Both types illustrated are effective in one direction only. The type of *Fig. 67b* is hardly suitable for use as a by-pass valve if the speed required is low in comparison with the maximum speed obtainable, since small variations in pump output will have a large effect under such conditions. Otherwise, it has the advantage (again when used as a by-pass valve) of leaving the main flow path absolutely unrestricted.

Metering valves of either type may be connected in the pump delivery main, just ahead of individual selectors, or in one or both lines between selector and cylinder, giving independent adjustment of speed in the two directions of motion. The type of *Fig. 67b* may also be used to throttle outlet flow from a selector or cylinder, in which case speed control is still retained even if the cylinder moves under negative load. The latter system is particularly useful for controlling the return speed of a single-acting cylinder.

A typical application of metering valves in machine

Fig. 67—Two types of by-pass metering valves

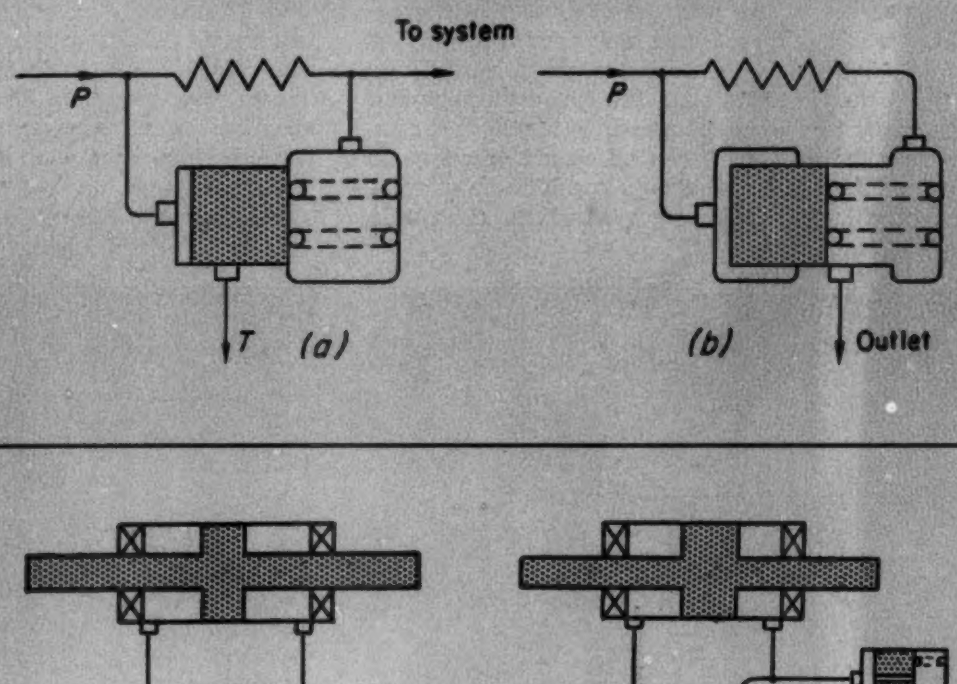
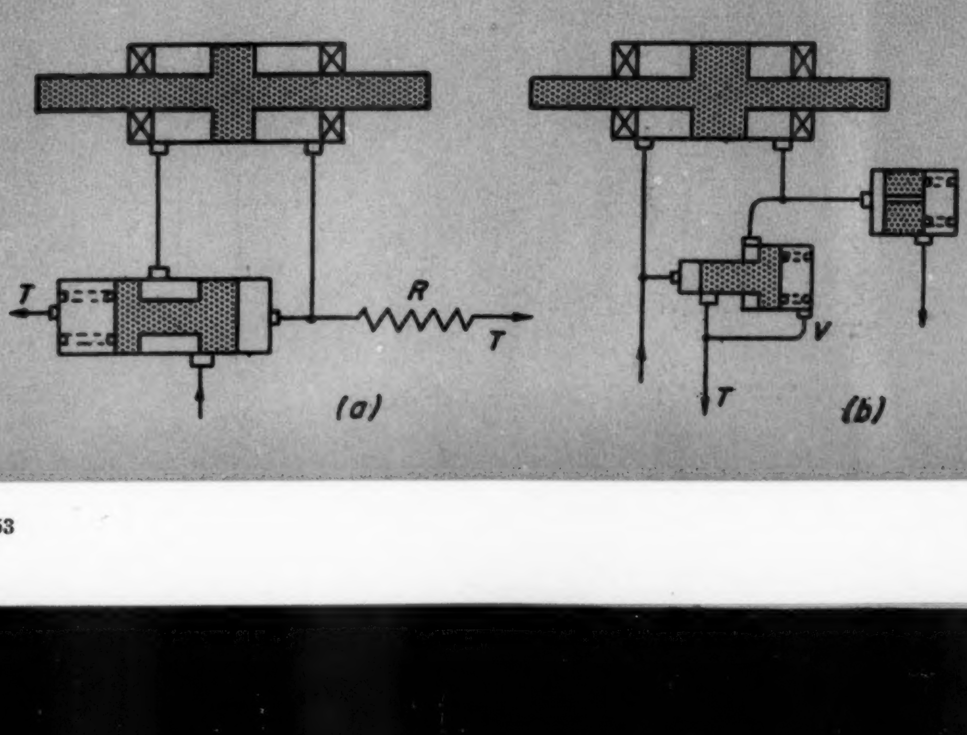


Fig. 68—One variation of the by-pass type of metering valve for speed control, *a*, and one for high negative loads, *b*



tools is to control the speed of slides. Frequently, as in production milling, there may be an "approach" portion of the slide travel, in which maximum speed is required, followed by a slow-speed working portion, followed in turn by a high-speed portion. For cases such as this, automatic control may be obtained by cam-operated valves arranged to alter the controlling resistance of the metering valve. Thus, in the case of the valves of *Fig.* 67*a* and *b*, if placed in the delivery line the valve is made ineffective by shunting the resistance. Again, if used in a bleed branch to tank, the valves can be made ineffective by closing the bleed branch. More generally, a cam-operated valve can be used to give any number of different values to the controlling resistance and thus give any number of speeds, or continuous speed variation. Different speeds in the two directions of motion are quite easily obtained by placing metering valves between the selector and cylinder.

Similar methods of controlling the speed over the stroke are applicable to two-pump systems by causing a cam-operated pilot valve to unload the larger pump when low speed is required. An example of this has been shown in *Fig.* 36. Again, the same principle is applicable with a variable-delivery pump if the output of the latter is governed by a resistance.

Speed control methods for machine tools have been developed to a high degree of refinement, particularly by the Cincinnati Milling Machine Co.[8] One possible variation of the by-pass type of metering valve is shown in *Fig.* 68*a*. Here the controlling resistance is in the return line, which helps in holding negative loads provided the latter are not too large. For high negative loads, a throttle type metering valve in the outlet line is used instead. With the latter arrangement, an improved type of circuit is shown in *Fig.* 68*b*. This circuit involves a special type of relief valve *V*, which is loaded by pressure in the return line as well as by pump pressure, the two acting preferably on equal areas. Thus, if the negative load corresponds to half the peak system pressure, the pump will relieve at half pressure, whereas with the simpler system the pump would have to be throttled down to full pressure. The result is that elastic deflection of the oil column on the pump side is halved, giving more accurate control of the slide, while power dissipation is also halved.

A different type of "by-pass" metering valve is shown in *Fig.* 69*a*. Here no springs are used and the valve works by bleeding off a fixed *proportion* of the pump output. Balance is reached when pressures on both sides are equal, i.e., when the pressure drop in R_1 is equal to that in R_3, or when $q_1/q_2 = R_2/R_1$ (for the sake of simplicity, resistances are regarded as linear but in practice of course they will probably follow a square law). Thus

$$\frac{q_1}{q_0 - q_1} = \frac{R_2}{R_1} \text{ or } \frac{q_1}{q_0} = \frac{R_2}{R_1 + R_2} \qquad (11)$$

a fraction $R_1/R_1 + R_2$ of the pump output is bled off to tank. It is pointed out[8] that it is desirable to adjust both resistances, decreasing R_1 and increasing R_2 to increase the flow to the system, thus avoiding excessive losses.

Fig. 69*b* shows another variation of the flow divider type of metering valve in which, as in Fig. 69*a*, one of the controlling resistances is in the return line. *Fig.* 70 shows another arrangement where the valve works by throttling the return flow and which is therefore only suitable for operation with negative loads. The circuits of *Fig.* 69 can only operate under positive loads. Circuits have been devised which combine both actions and thus give control under all conditions.

A somewhat different speed control problem arises

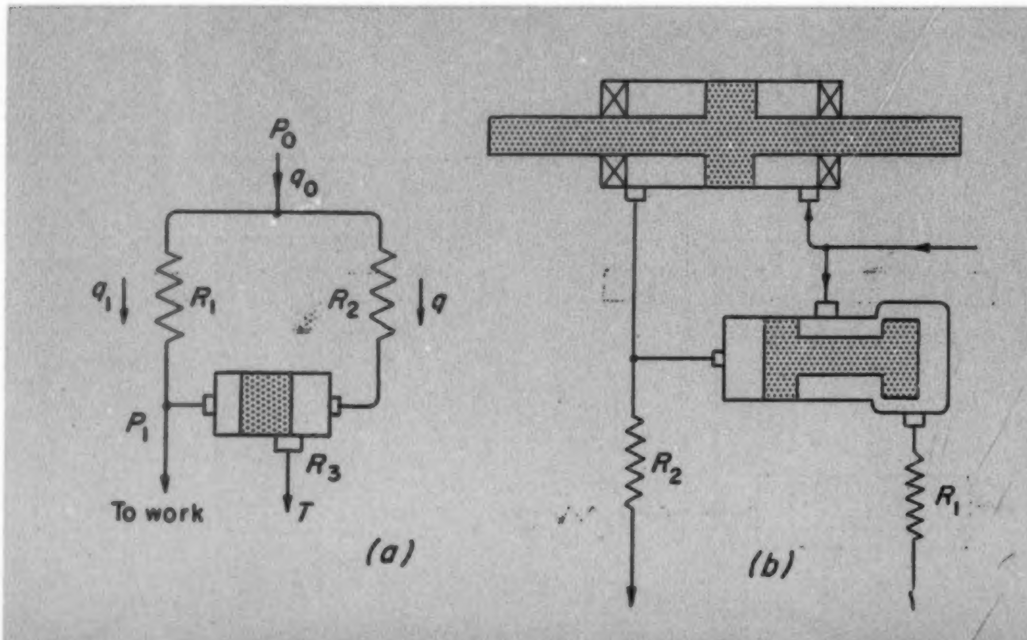


Fig. 69—Two types of "flow divider" metering valve arrangements for positive loads

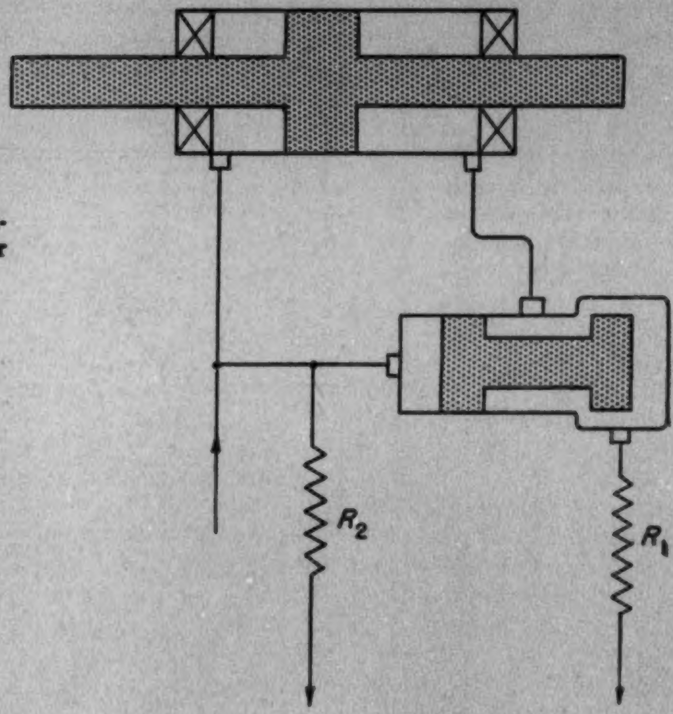


Fig. 70—Right—Flow divider metering circuit for negative loads

Fig. 71—Gradual type servo control valves can be designed for effective operation by utilizing tapered spools, *a*, or by using tapered grooves for low speeds, *b*

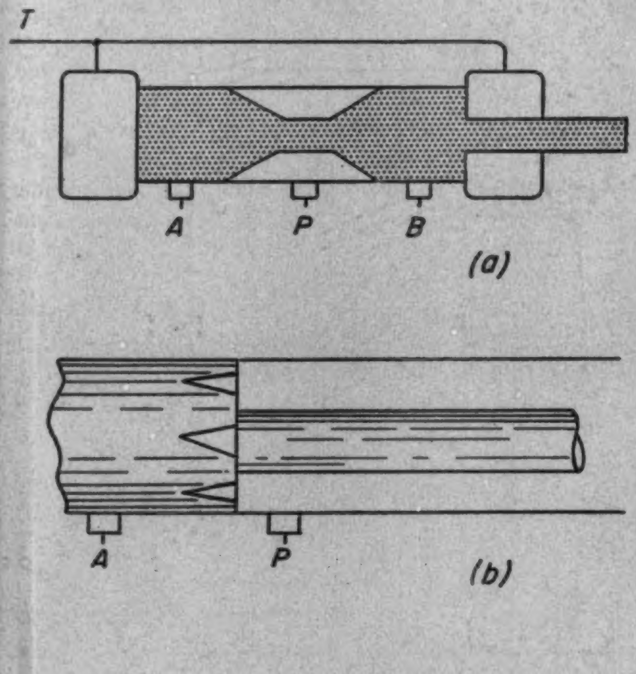
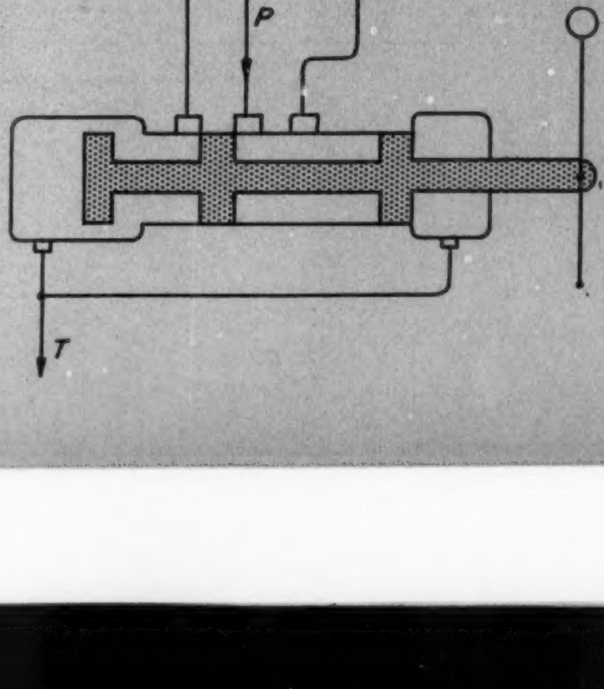


Fig. 72—Multiposition valve for obtaining two speeds



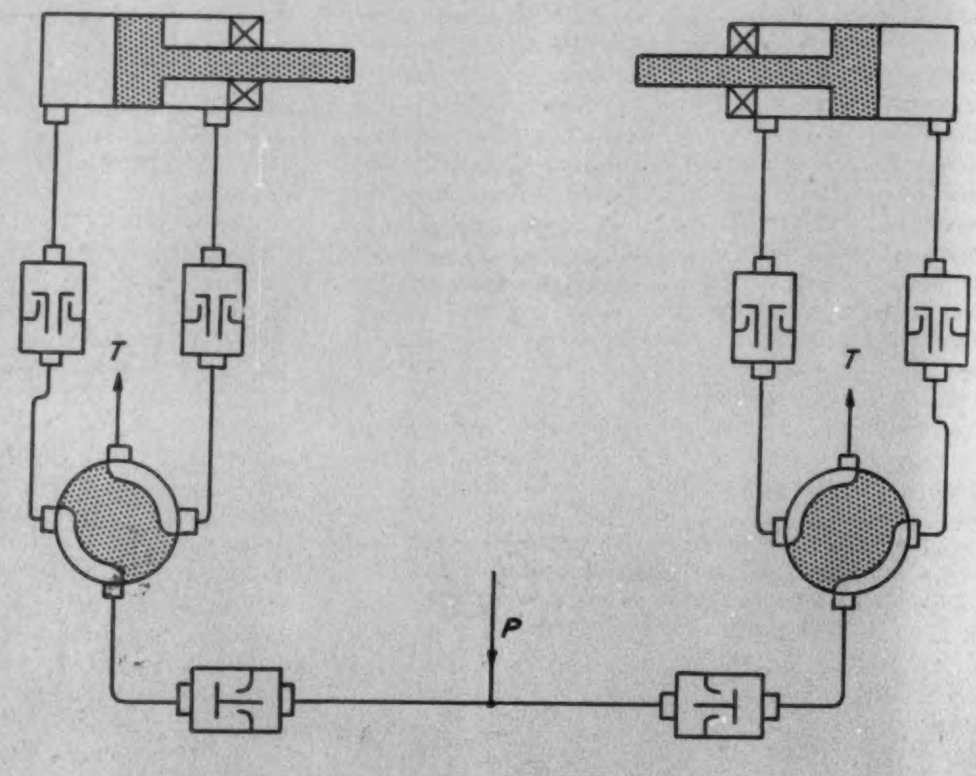
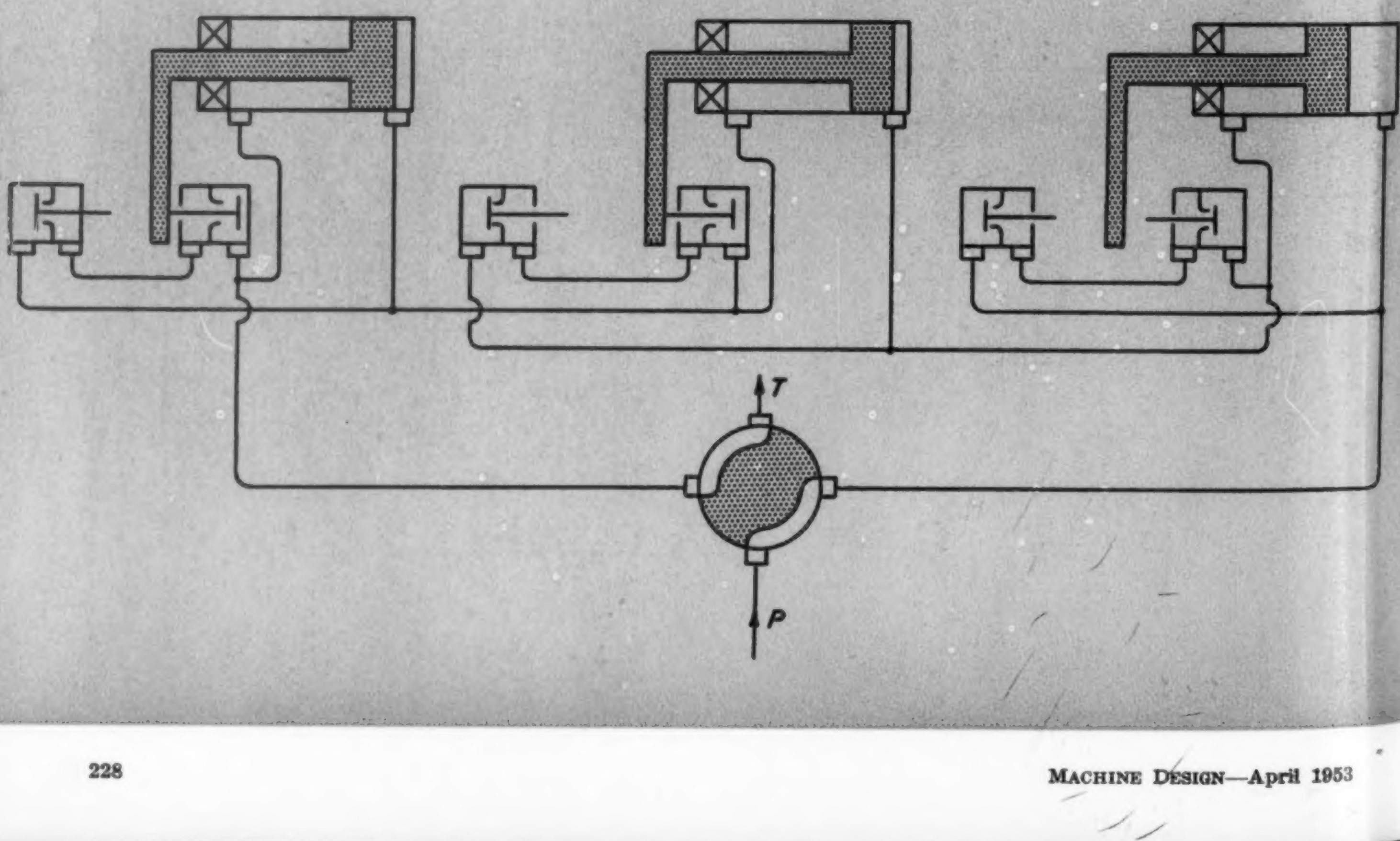


Fig. 73 — Circuit with two cylinders designed to operate together or independently

Fig. 74—Parallel operating cylinder circuit with thermal compensation

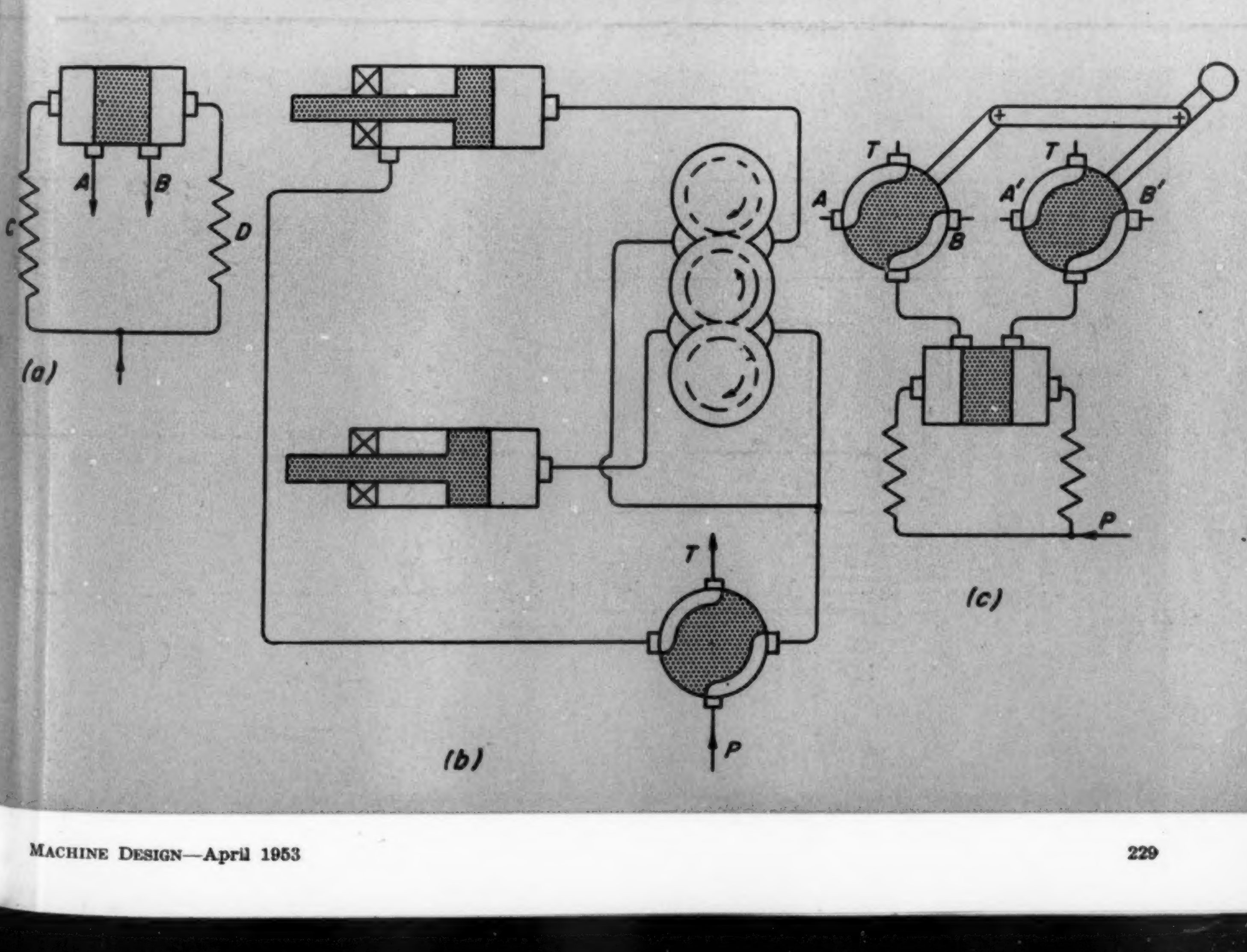


in applications such as handling machinery, gun turrets, and even certain presses (particularly when setting up) where it may be desirable to control speed according to the position of the selector lever to use the selector in a progressive manner, the speed increasing gradually as the control lever is moved further from the central position. Similar conditions arise in follow-up servo systems. Most selectors do not lend themselves well to this type of control unless specifically designed for it, as the opening of the passages is too rapid and effective control can only be achieved within a small range of motion near the central position, making the system far too sensitive. Gradual control can be achieved by using tapered spools, as shown in *Fig.* 71*a*, for slide valves. Similar principles apply to rotary and seating valves. Tapering may also be applied to the outlet side and to the ports as well as to the spool. If very low speeds are required, even tapered spools may be unsatisfactory, and a series of tapering grooves (as shown in *Fig.* 71*b*) is more effective.

In cases where only certain definite speeds are required (in practice two speeds only), one may also resort to methods similar to two-pump and variable-area systems, the latter being by far the simpler. Since the object now is to control speed for purposes other than power economy, such controlling methods are not suitable, with the exception of positional devices in the case of machine tool slides and similar applications. For manual control, it is possible to use multiposition selectors such as that of *Fig.* 72. In the left-hand position this gives high speed by causing the cylinder to operate "differentially", i.e., on the piston rod area. In the central position the cylinder advances on the full area and, therefore, slow speed is obtained. In the right-hand position the cylinder retracts. It is quite easy to devise similar selectors having four positions, one of which is neutral.

Synchronism: In certain applications it may be required that two or more cylinders without meat speeds bearing a given ratio to each other. An chanical interconnection move at the same speed, or example is found in certain aircraft landing flap systems. Without special provision, the cylinder having the smallest load would tend to move first, if two or more are connected in parallel. For the sake of simplicity this discussion will be confined

Fig. 75—Three methods of dividing flow to cylinders in parallel: by valve, *a*, gear motors *b*, or coupled selectors *c*



to the case of two cylinders, but the principles are applicable to any number.

A somewhat milder form of the problem is encountered in such applications as fork lift trucks, where hoist and tilt cylinders are expected to move simultaneously but exact synchronism is not demanded. In such cases, if a moderate amount of power wastage can be tolerated, a simple solution is obtained by inserting resistances in the lines to both cylinders, as shown in *Fig.* 73. The resistances are shown as being of the one-way type (restriction valves) to give independent adjustment in the two directions of motion, and as acting on outlet flow to give some measure of control under negative load. If q is the pump output; q_1 and q_2 the flows to the two branches; R_1 and R_2 the respective effective resistances (corrected to allow for differences between outlet and inlet flow); p_1 and p_2 the pressures required to move the two cylinders, assuming resistance to follow a square law; the relative cylinder speeds can be calculated from:

$$q_1 + q_2 = q$$

$$p_1 + R_1 q_1^2 = p_2 + R_2 q_2^2$$

This system applies to two or more cylinders which must be able to operate either together or independently. Accurate synchronism is usually required only in cases where the cylinders always operate together and two or more can then be connected in series. This, however, would allow synchronism to be upset by leakage or thermal expansion, for which there must be automatic compensations. Valves which open at the end of the stroke to short-circuit each cylinder can be used for compensation as shown in *Fig.* 74. The valves must allow free return and are usually mounted inside the pistons.

Because the foregoing system entails a number of difficulties it is seldom used. But, if two cylinders are in parallel, some means must be used to divide the flow between them equally or in a given ratio. *Fig.* 75*a* shows one form of valve for this purpose similar in principle to the metering valves described previously. The valve takes up a position depending on the ratio of flows through the two branches *A* and *B*. If flow in branch *A* rises at the expense of that in branch *B*, the pressure drop across the resistance *C* increases and that across *D* decreases, causing the valve to move to the left thereby throttling the flow to branch *A*, etc. For effective action, it would seem that proportions should be such that appreciable resistance to flow is offered even in the central position of the valve. In practice, the resistances *C* and *D* may consist of orifices within the valve spool through which flow takes place.

Another form of flow divider consists of two hydraulic motors, each connected in series in one of the two branches but mechanically coupled together and therefore constrained to rotate at the same speed. With gear motors a simplified form of this principle is possible, as illustrated in *Fig.* 75*b*, three gears being used.

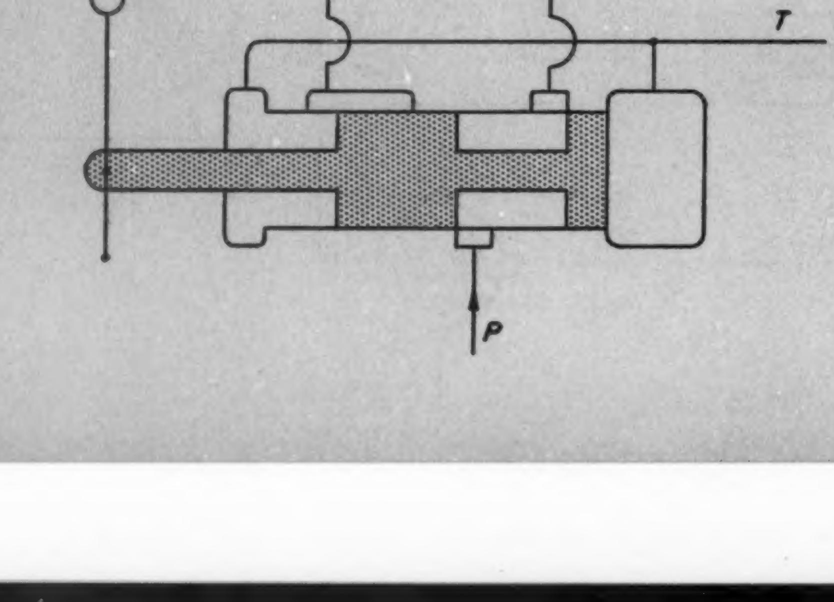


Fig. 76—Simplest type of three-position cylinder

As shown, the hydraulic motor type of flow divider connected between the selector and the cylinders will still act, when the latter are reversed, by controlling the outlet flow. The metering type of flow divider of *Fig.* 75*a* acts in one sense only and, if required to be effective in both directions, may be used in conjunction with two mechanically coupled selectors, as shown in *Fig.* 75*c*. It is also possible to devise metering type flow dividers which are effective in both directions.

When flow dividers are used, provision should be made to allow both cylinders to reach the end of their travel by allowing a small permanent leakage path to both branches.

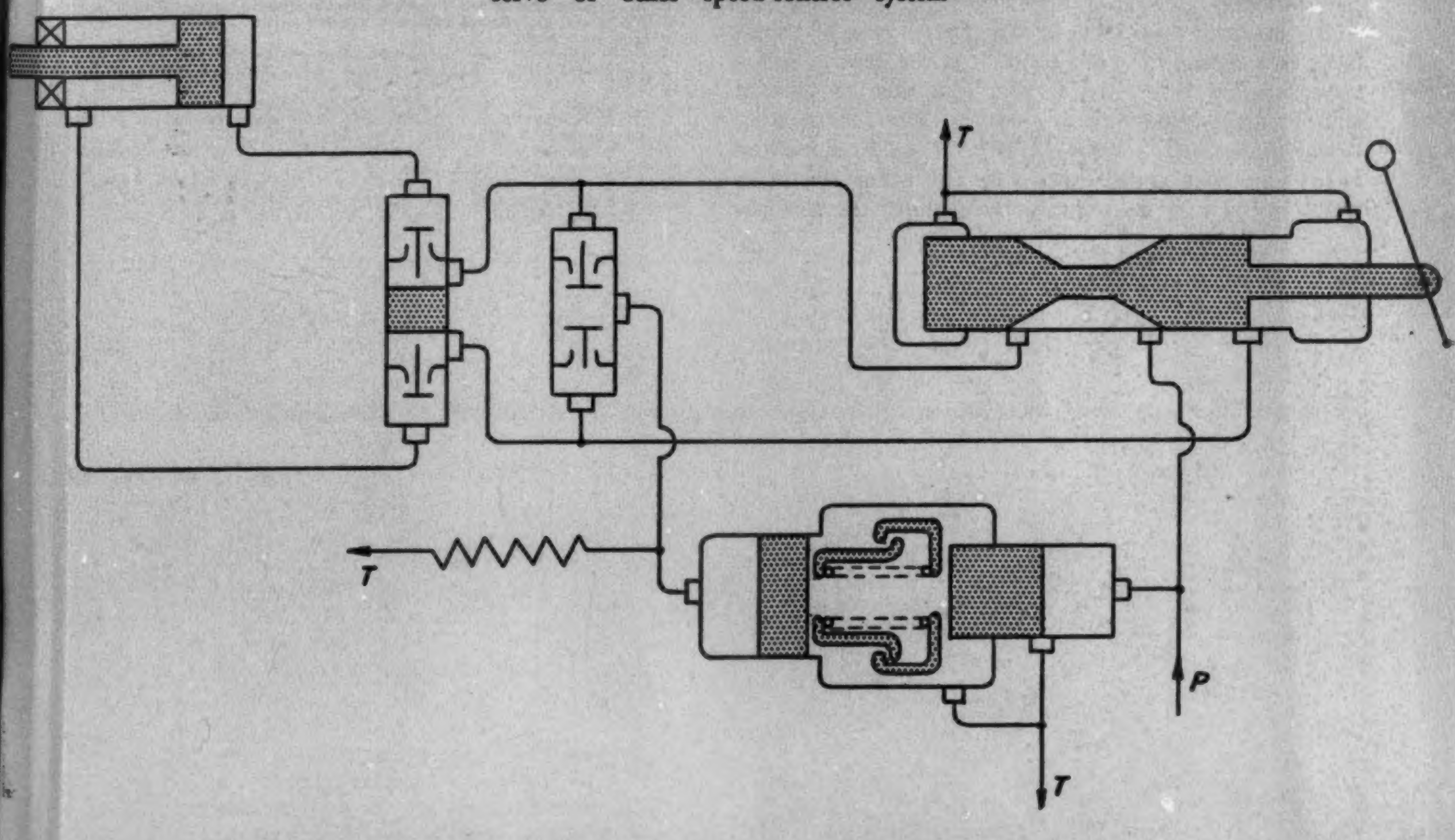
Multi-Position Systems: A typical case where a cylinder is required to occupy a number of different positions (greater than two) arises in aircraft landing flap systems, where three positions—"retracted", "take-off", and "landing"—are usually required. Another case arises with some automatic presses required to open wide when the work is being loaded or unloaded, but to operate over a small stroke while working in order to achieve the highest possible speed.

If the selector has a blocked neutral, an ordinary cylinder can always, in principle, be stopped in any intermediate position. Concern here is with systems in which one or more intermediate positions are preselected according to the position of the control lever or pushbuttons.

In such systems it is still possible to use an ordinary cylinder with positional sequence valves suitably interconnected with the selector in such a manner that the latter is automatically brought to neutral when required. This type of system can also be used (perhaps rather more conveniently) with electrical control. Suitable circuits for either case are easily devised.

In many cases the simplest solution is to use a multiposition cylinder with control gear to suit. For three positions and very low load, it is sufficient to use a spring-centered cylinder with a selector of the type which puts both lines to tank in neutral. With reasonably high loads, centering springs are out of

Fig. 77—Power economizing circuit for servo or other speed-control system



the question and other cylinder designs must be used. For more than three positions (or perhaps more than four) the cylinder and control gear are apt to become rather complicated and it may be preferable to use a follow-up servo system, even though infinite control of movement may not be required.

Probably the simplest type of three-position cylinder is that shown in *Fig.* 76. For the sake of simplifying the selector, this cylinder is often used differentially, connection *A* being permanently to pressure. With *B* to pressure (irrespective of the polarity of *C*) the cylinder is fully extended. With *C* to pressure and *B* to tank, the upper piston moves to the left against its stop and acts as a stop for the main piston which is urged to the right by pressure at *A*, an intermediate position thus being obtained. With *B* and *C* to tank the cylinder is fully retracted. The required combinations of connections could be obtained by an ordinary four-way selector of the type which puts both lines to tank in neutral, but this would give the three positions in the wrong order. Suitable selectors, however, are easily devised and an example is shown in the diagram. There are several other types of three-position cylinders, usually requiring different types of selectors.[1]

Special Power Economy Problems: When the output of a constant-delivery pump is being throttled, the pump must work at peak pressure for the throttling to be effective even though the pressure required may be quite low. Hence serious power wastage may be entailed.

The problem may arise in the general case of throttling as a means of speed control, in the case of selector-controlled follow-up servos, and also in certain circuits embodying reducing valves. For economizing power under such conditions, *Fig.* 77 shows a method as applied to a servo system or any other circuit in which speed may be reduced by opening the selector partially. The system is based on a special type of relief valve, similar to the combined relief-unloading valve of *Fig.* 6, but the upper or "loading" piston is only a little larger than the valve spool. Interposed between the loading piston is a spring in a preloading cage which only comes into operation when peak pressure is reached, in which case the upper piston travels down to a stop, and the device fuctions as an ordinary relief valve.

Below peak pressure the spring merely acts as a rigid distance piece. The upper piston is loaded by pressure from beyond the selector or other throttling device. As long as there is flow and throttling, the valve will automatically set itself at a pressure only slightly higher than that required to move the cylinder, the difference being lost by throttling. If the cylinder load is of the nonreactive kind (friction or equivalent) the pump will be unloaded when the selector is brought back to neutral, pressure on the upper piston being allowed to leak out through the restrictor. With a reactive or "live" load the same effect can still be obtained by interposing a hydraulic lock between the cylinder and the branches leading to the nonreturn valves, as shown.

Further basic analyses of machine hydraulics circuits will be discussed in subsequent issues of MACHINE DESIGN. These will be devoted to separate coverage of four specific types: open-center, pilot and electrical control, automatic sequence, and servo systems.

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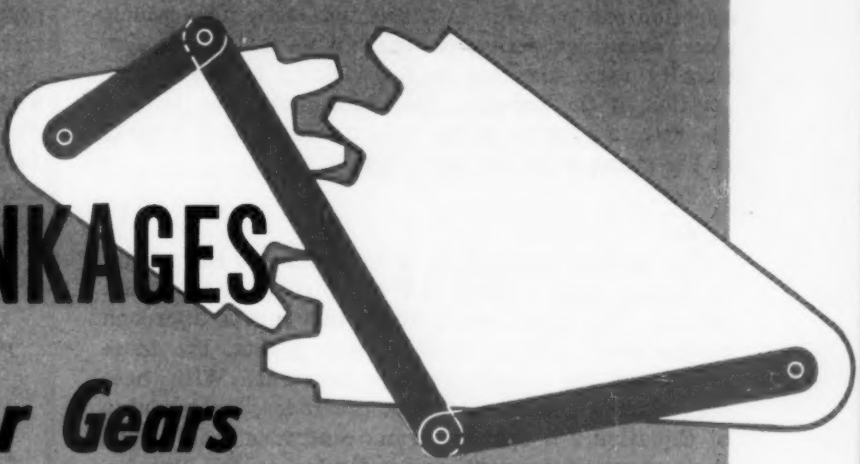

Designing FOUR-BAR LINKAGES *to Substitute for Gears*

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FOR transmission of a substantially constant angular velocity ratio for a limited range of angular motion, the four-bar linkage may be a simple and effective substitute for more expensive toothed gearing. A suggested procedure for designing such linkages is presented in this article. Also, sample designs and velocity ratio curves are presented.

The linkage in *Fig.* 1*a* transmits a velocity ratio

$$\frac{\omega_{in}}{\omega_{out}} = n = \frac{OP}{O'P} = -2 \quad \text{(1)}$$

where O' and O are the pivots of the input and output cranks, respectively, and P is the common instant center between the cranks in the position shown. In general the velocity ratio will be different from -2 for any other position of the same linkage. This may be seen from the graph of velocity ratio versus high-speed crank angle, *Fig.* 1*b*. Obviously, if a nearly constant velocity ratio of -2 over an appreciable range of motion is desired, this particular design is poor since the ratio is close to -2 for only a limited range of crank rotation.

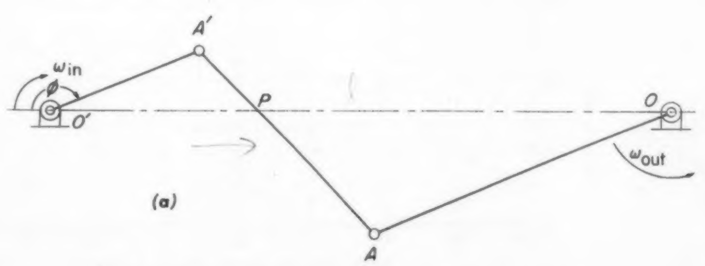
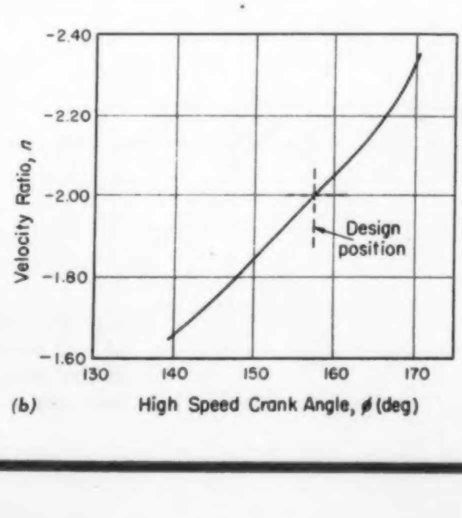


Fig. 1—Four-bar linkage, *a,* designed for a velocity ratio, $n = \omega_{in}/\omega_{out}$, of -2 in the position shown. Velocity ratio plotted at *b,* varies greatly with crank position



Dimensions of the linkage in *Fig.* 1*a* were selected at random. A better design can be expected if the selection can be made in accordance with rules which place some restriction on the shape of the angular velocity ratio diagram in the vicinity of the design position of the linkage. One such restriction is that the rate of change of the velocity ratio with respect to the crank angle be zero; that is,

$$\frac{dn}{d\phi} = 0 \quad \text{(2)}$$

The motion transmitted is equivalent to the rolling of two curves in contact at P and pivoted at the crank pivots. The more nearly these two curves approach circles with centers at the crank pivots, the more nearly will the velocity ratio be constant. With these curves identified as the *crank centrodes,* the meaning of Equation 2 is that the crank centrodes should have their common tangent perpendicular to the line between crank pivots. The linkage shown in *Fig.* 2*a* was designed in accordance with this rule. The curve of *Fig.* 2*b* shows that there is a definite, though short, range of motion in which the velocity ratio is essentially constant and hence the design is an improvement over that of *Fig.* 1.

The design may be further improved by use of the relationship,

$$\frac{d^2n}{d\phi^2} = \frac{dn}{d\phi} = 0 \quad \text{(3)}$$

This criterion means so designing the linkage that the crank centrodes not only have their common tangent perpendicular to the line of pivots but also have centers of curvature at the pivots, and hence more closely approach the shape of circles centered on the pivots. The linkage of *Fig.* 3*a* is so designed. The diagram of *Fig.* 3*b* shows that the range of constant velocity ratio (within $\pm$ 1 per cent) has been considerably extended and might be quite adequate for many applications.

Design Details: The following design equations implement the process described:

$$PO = \frac{ln}{n-1} \quad \text{(4)}$$

$$PO' = \frac{l}{n-1} \quad \text{(5)}$$

$$PQ = \frac{3ln}{(n-1)(2-n)} \quad \text{(6)}$$

$$PQ' = \frac{-3ln}{(n-1)(1-2n)} \quad \text{(7)}$$

All distances are directed, and the positive sense is from O' to O. The symbol l stands for the distance

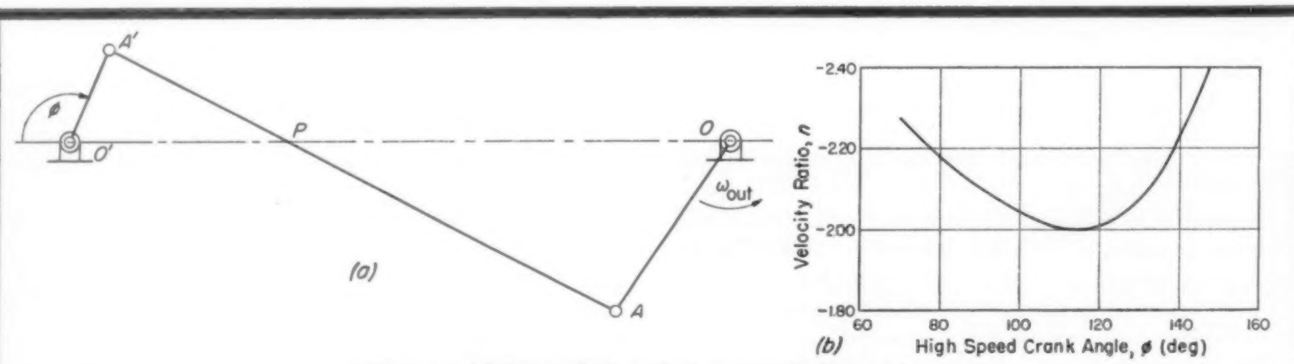


Fig. 2—Linkage designed for a velocity ratio of -2 under the restriction that $dn/d\phi = 0$ in the design position. Velocity ratio, *b,* is nearly constant over a very small range of crank motion

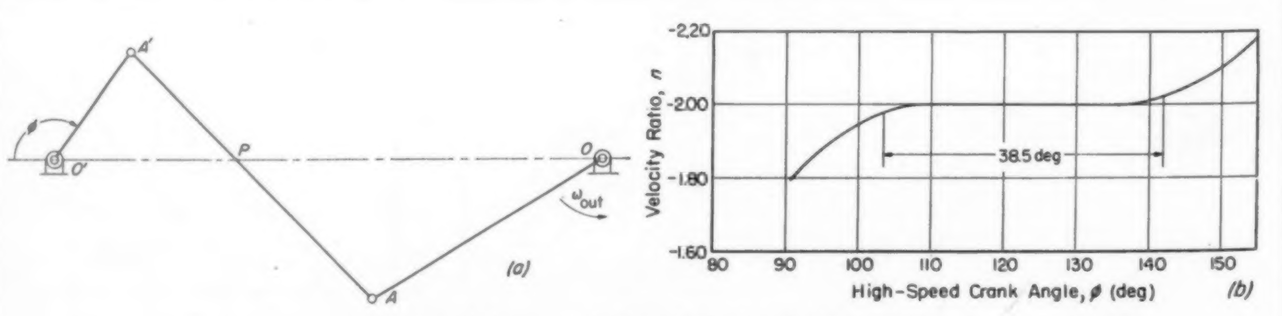


Fig. 3—Linkage, *a,* designed for a velocity ratio of -2 under the restriction $d^2n/d\phi^2 = dn/d\phi = 0$ in the design position. Velocity ratio, *b,* is nearly constant over a substantial range of crank motion

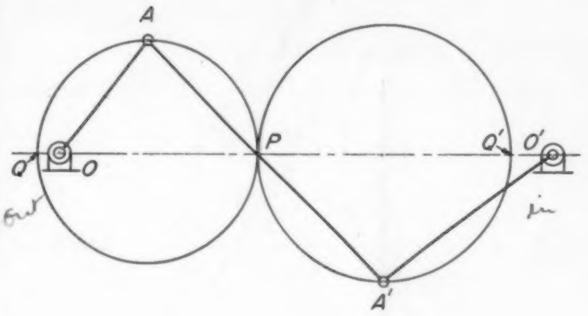


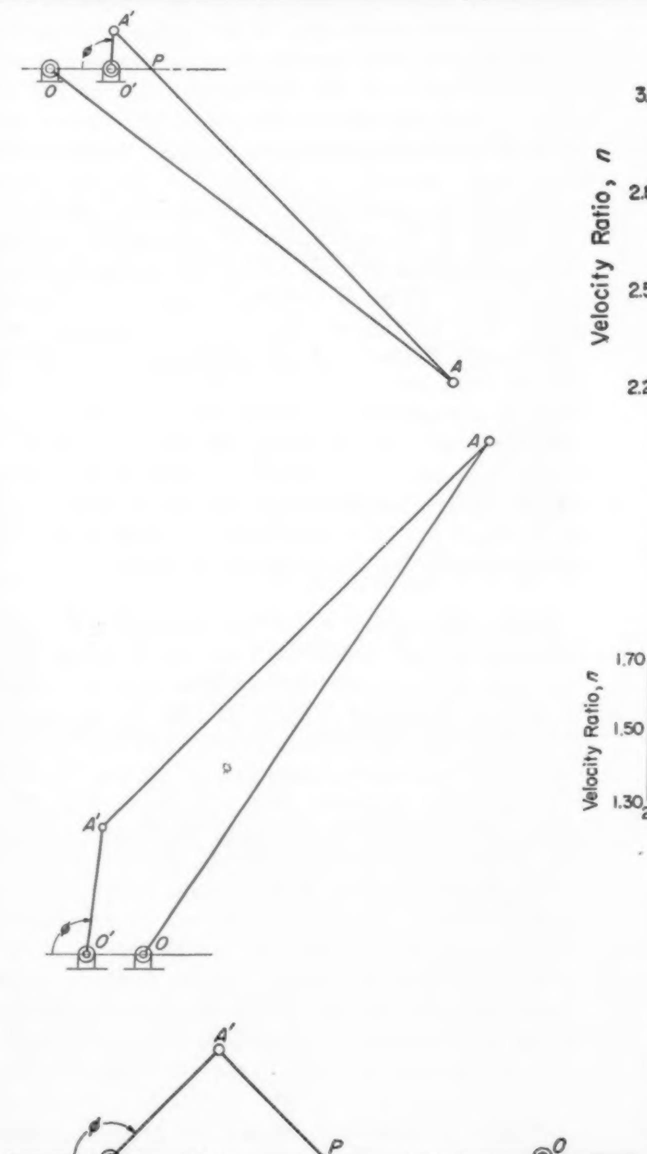
Fig. 4 — Design layout for a velocity ratio of $-2/3$

$O'O$. The angular velocity ratio, n, is positive if the two cranks rotate in the same sense.

Suppose that a design is desired for a velocity ratio $n = -2/3$ and a distance between pivots of $l = 6$ inches. From Equations 4 through 7,

$$PO = \frac{6\left(-\frac{2}{3}\right)}{-\frac{2}{3} - 1} = 2.40 \text{ inches}$$

$$PO' = \frac{6}{-\frac{2}{3} - 1} = -3.60 \text{ inches}$$



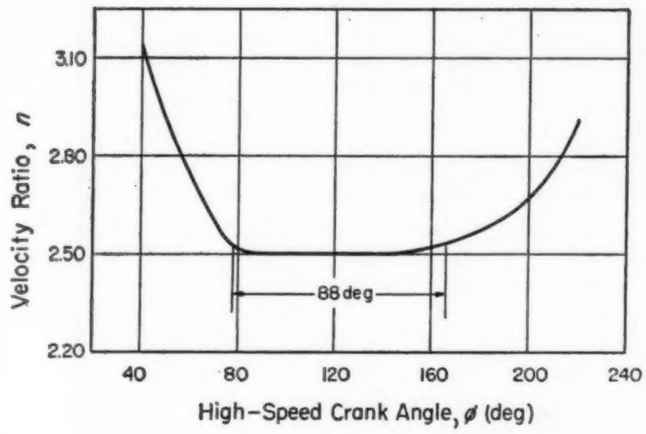
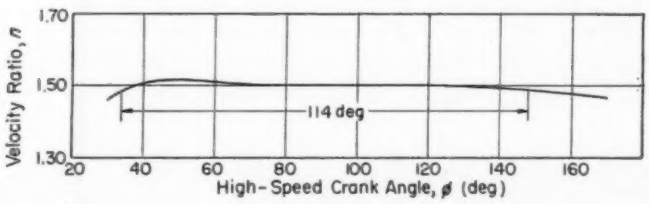
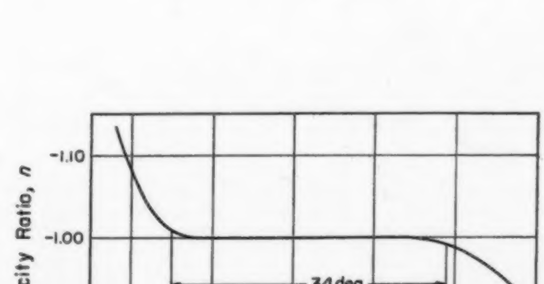


Fig. 5—Designs for velocity ratios of 2.5, 1.5 and -1.0, and corresponding velocity ratio graphs. These are sample designs, not necessarily the best





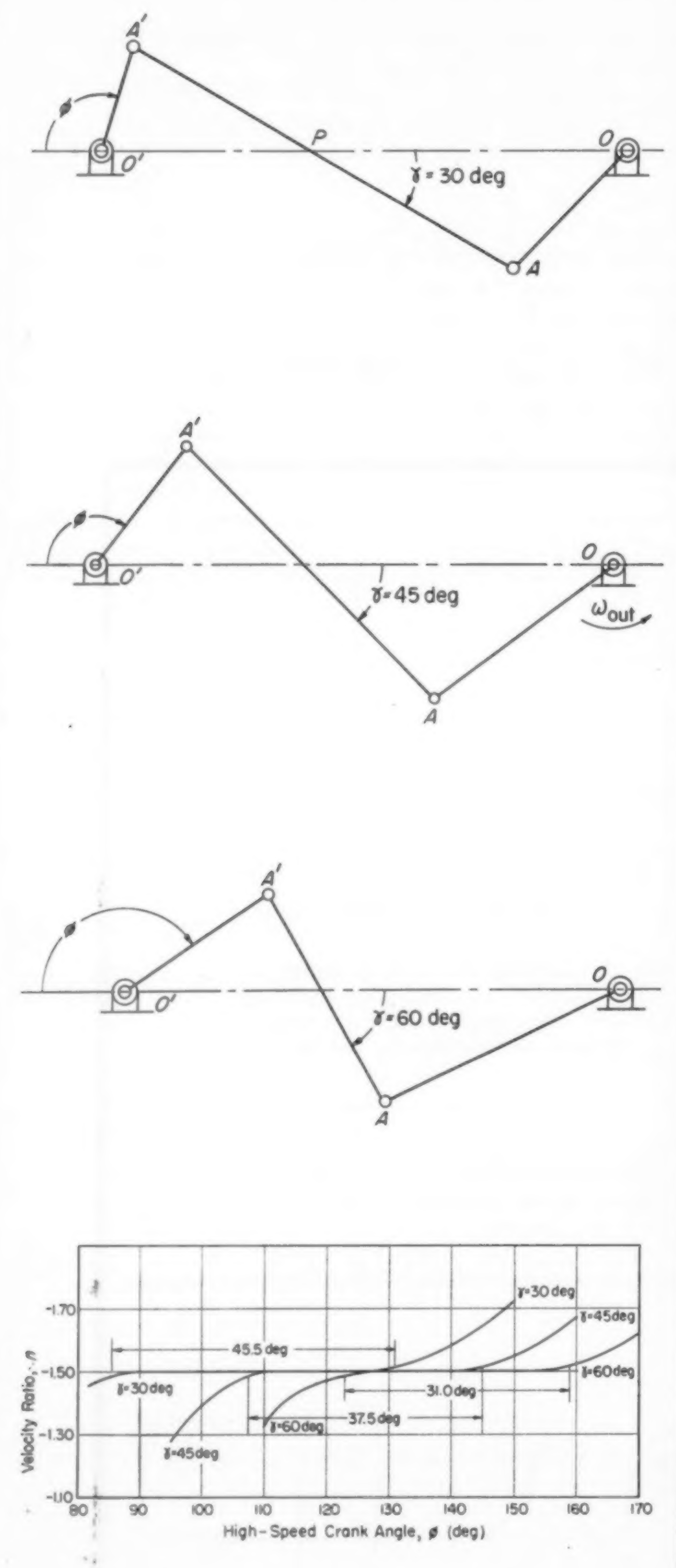


Fig. 6—Differentiated by angle γ, three different designs for a velocity ratio of -1.50, and velocity ratio curves. For a velocity ratio of -1.50, small values γ yield best designs

$$PQ = \frac{3\,(6)\left(-\frac{2}{3}\right)}{\left(-\frac{2}{3}-1\right)\left(2+\frac{2}{3}\right)} = 2.70 \text{ inches}$$

$$PQ' = \frac{-3\,(6)\left(-\frac{2}{3}\right)}{\left(-\frac{2}{3}-1\right)\left(1+\frac{4}{3}\right)} = -3.09 \text{ inches}$$

These distances are marked on a layout as shown in *Fig.* 4. Then, with *PQ* and *PQ'* as diameters, two circles are drawn. Crankpin *A* can be selected anywhere on the *PQ* circle and the corresponding location of *A'* found on the *PQ'* circle by drawing a straight line (the connecting rod) through *A* and *P*. Nothing in the design procedure indicates what choice of location for *A* will give the best results, but experience has shown that locating *A* at the intersection of the *PQ* circle with a perpendicular to *OO'* from the center of the circle gives consistently good results. After *A* and *A'* have been tentatively chosen, the linkage may be drawn and its velocity ratio plotted as a check.

A point to be emphasized in this design process is that restrictions are placed on the linkage at one position only. There is nothing in the process to guarantee that the speed ratio will be constant within satisfactory limits over any particular range of crank motion. Whether or not a tentative design is satisfactory can only be determined by plotting the actual velocity ratio over the range of motion desired. The design process does eliminate a multitude of designs which are not worth trying, leaving a smaller group to be tested to discover whether a linkage exists which is satisfactory for the purpose in mind.

Other Examples: A further understanding of the possibilities and limitations of the four-bar linkage for transmitting a constant angular velocity ratio may be gained from the designs in *Fig.* 5. In each case the foregoing process was followed in arriving at the design and crankpin *A* was located on the *PQ* circle at one end of the diameter perpendicular to *OO'*. The designs are not necessarily those which give the greatest range of motion with the design velocity ratio.

The designs in *Fig.* 6 were prepared to show, for a design velocity ratio of -1.50, the effect of different locations on the *PQ* and *PQ'* circles for the crankpins *A* and *A'*. The parameter γ is the angle between the connecting rod and the line of pivots in the design position of the linkage. For this example the trend indicates that small values of γ give the best results.

"With professional management and technology America's future is assured. . . . Unless engineers and engineering educators seize and answer this call to leadership, others will grasp and exploit it. Then the engineering profession, engineering education, and industry will have missed a great opportunity." —HARRY RUBEY, *professor and chairman, Department of Civil Engineering, University of Missouri.*

HIGH-PRESSURE PNEUMATICS

Design considerations for maximum efficiency in energy storage

By James L. Dooley
*Engineering Manager
Rhodes Lewis Co.
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PERHAPS the greatest single factor distinguishing pneumatic systems from mechanical, hydraulic or electric methods of doing the same job is the facility to store large quantities of energy as compressed gas available for instant use without complicated energy conversion equipment. Pneumatic energy resembles a flywheel in this respect. In fact, high-pressure pneumatic systems generally cannot be justified unless the requirements for high peak power of short duration or minimum weight exist. About the only exception is where high-density gas is required for chemical purposes. For continuous-duty loads, other methods are generally more efficient. Because of this the characteristics of pneumatic energy storage and associated equipment should be examined closely to completely exploit the advantages.

Pneumatic Energy: First, examine the paradoxical energy flow in a pneumatic system. The compressor requires power to operate but practically all of this energy is disposed of as heat loss in the compressor cylinders themselves, the aftercooler, and the compressed air receiver. When the compressed air in the receiver reaches the same temperature as the compressor inlet air, the energy of compression is completely removed. But where does the energy come from to make the compressed air do work such as run an air motor or operate a piston in a cylinder? It comes from the internal energy in the air itself. Removal of this internal energy manifests itself as cooling of the air being used when work is being done with it, i.e., the compressed gas removes heat from itself by lowering its temperature to do the work required. This reduction in gas temperature results in a reduction of the availability of the energy in the remaining gas in the receiver.

Pressure, P, temperature, T, and volume, V, rela-

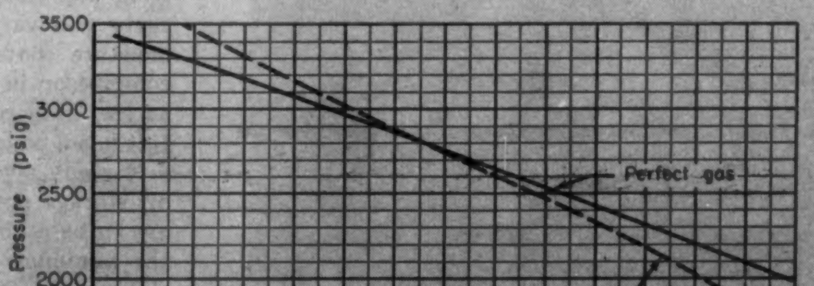


Fig. 1—Comparison of actual air to theoretical gas under temperature variation. Sealed receiver, charged at 70 F to 2800 psi was subjected to temperature changes encountered in aircraft

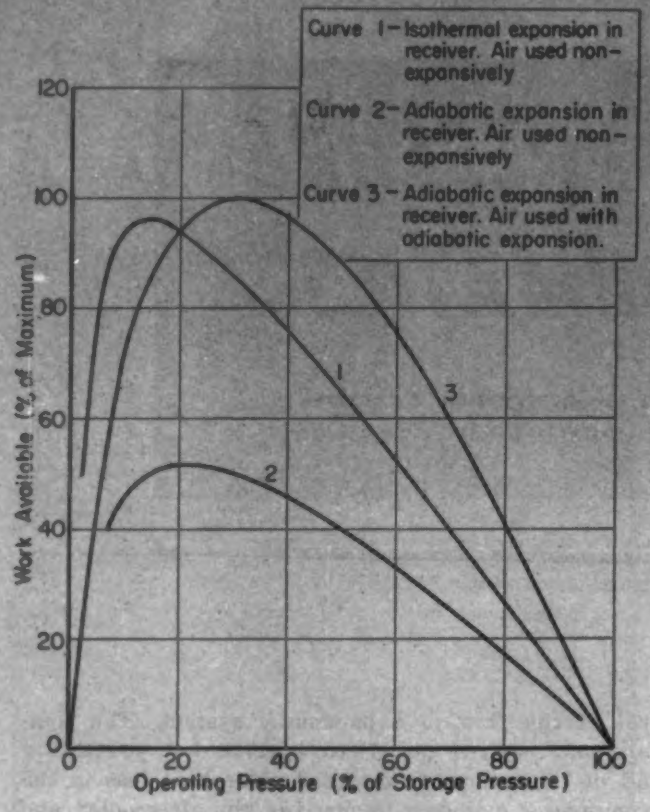


Fig. 2—Left—Variation in energy available for different operating pressure ratios

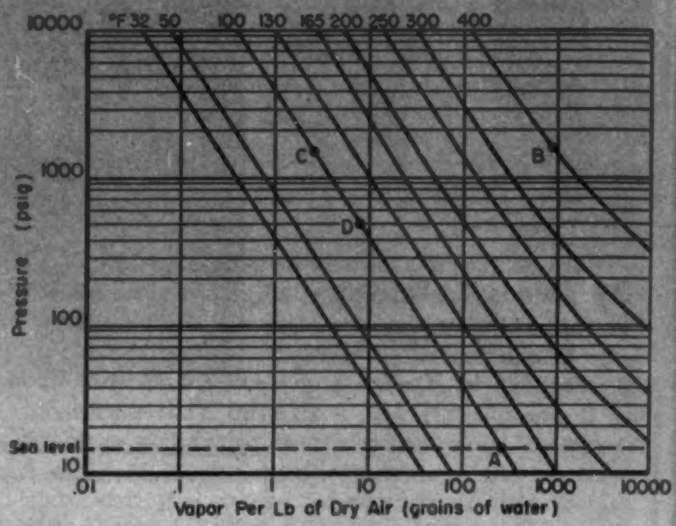


Fig. 3—Above—Water remaining in compressed air as vapor under various pressure and temperature conditions. Chart is based on theoretical gas and the partial pressures

tionships can be conveniently expressed for either expansion or compression (adiabatic) of a perfect gas by the basic gas laws which are given here for ready reference. Subscripts 1 and 2 denote the before and after conditions.

$$\frac{P_2}{P_1} = \left(\frac{V_1}{V_2}\right)^{1.395} \qquad (1)$$

$$\frac{V_1}{V_2} = \left(\frac{P_2}{P_1}\right)^{0.717} \qquad (2)$$

$$\frac{V_1}{V_2} = \left(\frac{T_2}{T_1}\right)^{2.534} \qquad (3)$$

$$\frac{T_2}{T_1} = \left(\frac{V_1}{V_2}\right)^{0.395} \qquad (4)$$

Nomenclature

P = Absolute pressure, pounds per square foot
V = Volume of gas, cubic feet
p = Absolute pressure, psi
v = Volume of gas, cubic inches
W = Weight of gas, pounds
R = Gas constant for air = 53.3
T = Absolute temperature, degrees Rankine (F + 460)

$$\frac{P_2}{P_1} = \left(\frac{T_2}{T_1}\right)^{3.534} \qquad (5)$$

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{0.283} \qquad (6)$$

Actually, gases in general and air in particular do not exactly obey these "perfect gas" relationships. In certain applications where large temperature changes are encountered and the total energy storage is important (as in aircraft) this variation must be taken into account. *Fig.* 1 shows actual measurements taken by varying the ambient temperature on a sealed air receiver and noting the pressure variation therein. The theoretical variation as computed from Equation 5 is plotted for comparison. This curve will vary slightly depending largely upon the moisture content of the air. In actual use, with a compressor in the system, the receiver is recharged as the temperature and pressure drop, hence this effect is not so marked.

For any given job to be done there is an optimum air storage pressure. This depends upon what factors are to be optimized. In many industrial installations the minimum installation and operating cost is desired while space and weight are of almost no importance. To handle a given job, low-pressure storage may be used but larger air receivers and compressors may be required than if a higher pressure is selected. Hand-held pneumatic tools, for example, become larger and heavier at the lower pressures, making the operation less efficient. Although each

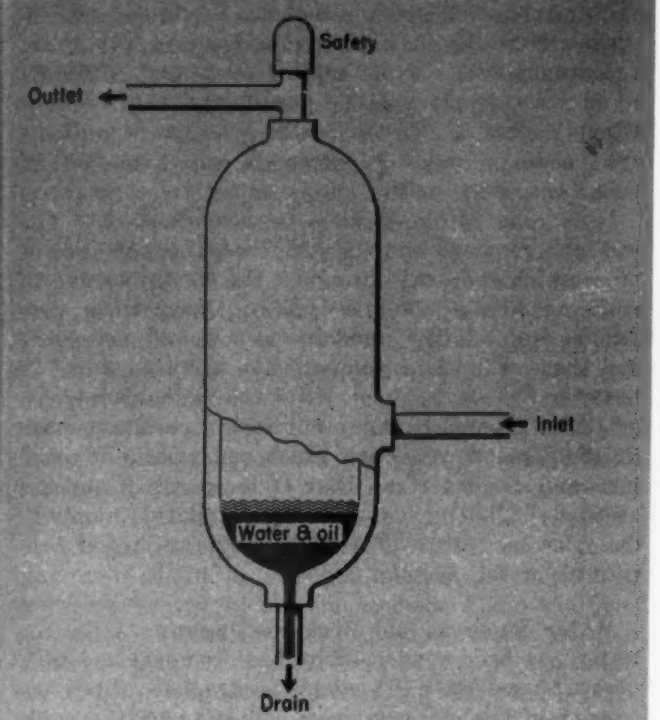


Fig. 4—Schematic cross section of typical air receiver designed to minimize carrying over entrained water and oil

individual case must be analyzed in the light of factors that apply primarily to it, initial cost, installation and maintenance must be considered not only for the receiver itself, but for the complete system.

As in a flywheel, where energy can only be removed by reduction in speed, the air receiver will give up energy only when a pressure reduction occurs. There the similarity ends, however, because in the gas secondary effects take place. Pressure reduction results in temperature reduction unless the change occurs so slowly that heat can enter to maintain isothermal conditions. This temperature reduction reduces the availability of the remaining energy. Because of this, together with the fact that the exhaust air must be dumped against atmospheric pressure, the actual optimum operating pressure varies with the method of use, the rate of use and the storage pressure as shown in *Fig.* 2.

Condensation Problems: In almost all air storage systems there exists the problem of condensation of water and oil from the compressed air with attendant problems of removal, corrosion and freezing. In some high-pressure systems the removal of portions of this water and oil becomes mandatory to avoid malfunctioning. This is especially true for use in aircraft or in systems exposed to low-temperature operating conditions. *Fig.* 3 shows how much water can remain in the compressed air as vapor under various pressure and temperature conditions. It is a theoretical plot based on the laws of partial pressures.

It is helpful to examine what happens to moisture-laden air as it proceeds through a high-pressure pneumatic system. Point *A* of *Fig.* 3 shows that at 100 F and sea level atmospheric conditions, one pound of compressed air will contain 270 grains of water vapor under saturated conditions. One grain of water is a large drop (¼-inch in diameter) and one pound of air is approximately 14 cubic feet. If this air, compressed to say 1500 psig, leaves the compressor at 400 F, it could contain at point *B* 900 grains of water as vapor per pound of air. However, since this additional water is not available the relative humidity at this point is 30 per cent.

As this compressed air is cooled in an aftercooler, in the lines, or in the receiver, point *C* will be approached and under these circumstances the air can contain only 3 grains of moisture per pound of air. Since this is less than was taken into the ma-

Fig. 5—Forged and welded stainless steel spherical receiver designed to hold 200 gallons at 5500 psi working pressure at −340 F. Used by the U. S. Air Force in rocket propellant experiments, the unit has a high safety tolerance and a pressure vessel index of 50,000. Nearly one ton of welding rod was used and each weld pass Dy-Cheked for porosity. Courtesy Turco Products, Inc.

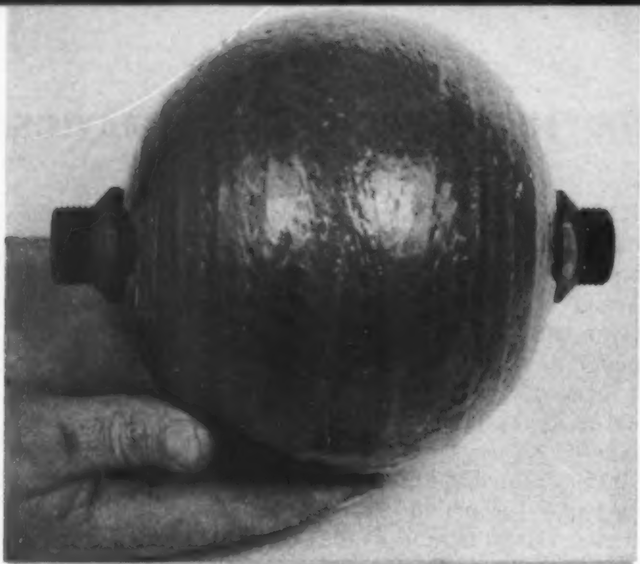


Fig. 6 — Above — Fiber glass wound, plastic-impregnated spherical air receiver for 3000 psi working pressure with volume of 113 cubic inches and weight of two pounds

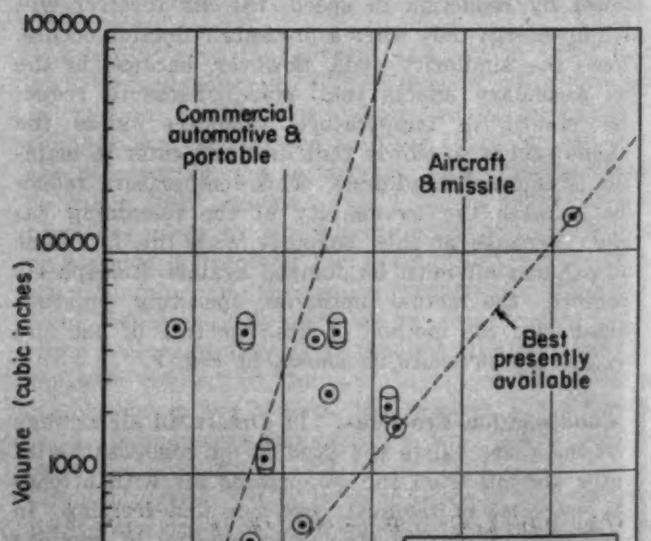
Fig. 7—Below—Typical commercial installation of a high-pressure pneumatic system. Receiver is used at 3125 psi, holds 3500 cubic inches

chine, obviously the excess water must condense out and be collected somewhere as liquid. As is obvious from *Fig.* 3, if the temperature is further reduced more water vapor will be condensed, although in an increasingly smaller percentage. Under conditions below 32 F, condensate will freeze and, as can be seen, only an extremely small amount of water can exist in the air as vapor.

It is also of importance to note that once the water is removed and the air is used at some lower pressure than storage pressure, the conditions shown as *D* on *Fig.* 3 may be typical. Under these conditions the relative humidity is low (38 per cent) and there will be no condensation at this point. If, however, work is done with the compressed air, adiabatic expansion may occur with a resultant drastic temperature reduction which condenses out moisture and freezes it so that it temporarily appears as snow. Under conditions of low relative humidity the flowing air will actually remove water entrained in the system.

Water Removal and Receiver Design: After the water has been condensed by heat removal and temperature reduction it exists as droplets either entrained in the air system or collecting on the walls and in low spots. This water should be collected

Fig. 8—Volume and available work per pound of receiver for varying working pressures. Note the small change in available energy. This analysis is based on spherical steel shells at 30,000 psi stress with isothermal receiver expansion and the air used nonexpansively at one-quarter of receiver pressure

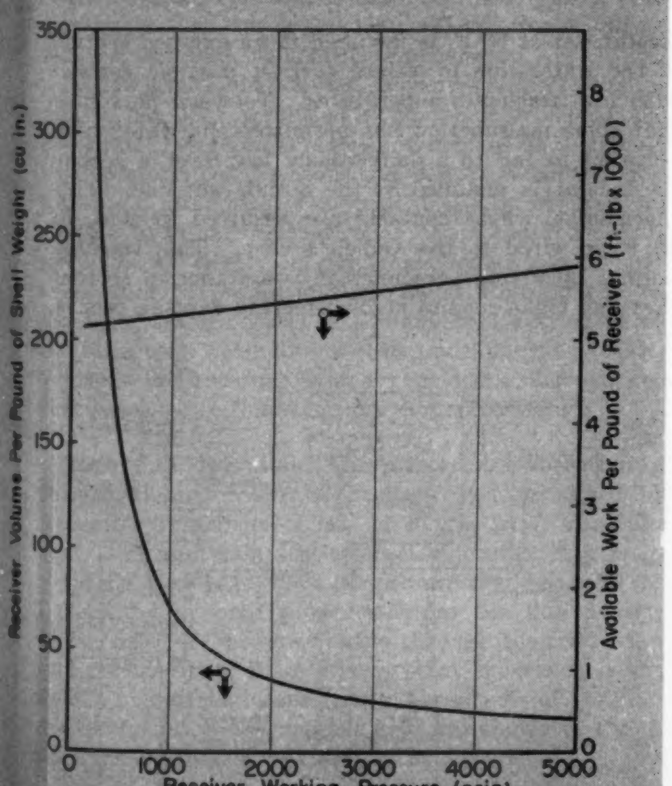


and removed, one very good place to do this being in the receiver proper. In the receiver the air is relatively stationary and the entrained moisture settles so it may be drained if the receiver is properly piped, *Fig.* 4. This separation may be aided by spinning the incoming air to effect centrifugal separation or by passing the air through chemically treated filter materials that reduce the surface tension thereby improving collection.

The problem as to the optimum shape for a high-pressure vessel to do a given job has been a subject of much discussion, especially where minimum weight and space both must be given consideration. It has been quite well established that the long cylindrical drawn tube with the ends swaged down is the most economical for general applications, although there are several noteworthy exceptions to this. One, shown in *Fig.* 5, is a very heavy welded spherical shell for 5500 psi service.

Although spherical shells are more difficult to fabricate and install in cramped places, they are used wherever weight is a prime consideration; classical stress analysis showing the hoop stress to be one-half the longitudinal stress explains this. In most spherical designs the joint falls on the great circles hence the point of highest stress has the weakest, most questionable section unless considerable reinforcement is used. This difficulty may be avoided by wrapping the sphere with high-tensile wire or other material. *Fig.* 6 shows an experimental glass fiber wound plastic-impregnated sphere with an inner sealing lining. This unit weighs less than the present best steel shells.

Fig. 9—Pressure vessel index versus receiver size showing what can be expected from current manufacturing practice



In designing air receivers, shattering on impact must be considered. The high specific energy storage causes the entire receiver to shatter and the pieces become shrapnel. Experience has indicated that steel cannot be heat treated to much above 120,000 psi yield point if shattering is to be avoided. Because of this, together with possible malfabrication, the effects of corrosion, embrittlement, and rough handling, the design and manufacture of high-pressure vessels is closely controlled by the codes of standard practice. *Fig.* 7 is a typical unit built under commercial codes.

Receiver Size: In selecting the size of receiver to handle a given job it must first be established just how much air is required from the receiver before the recharge compressors have time to become effective, even if the recharge system is in operation most air withdrawals occur so rapidly the amount of air added during this time is negligible. Where only one or two repeated cycles of operation occur and the total pressure-ratio change is not too great, it is convenient to make an approximation on an air-weight basis. From the basic gas laws

$$W = \frac{PV}{RT}$$

$$= \frac{pv}{340{,}000} \qquad (7)$$

For example, to ascertain the receiver size required to make one operation of a 300 cubic inch displacement cylinder at 1700 psi, minimum, when the initial receiver pressure is 2250 psi, the cylinder requires

$$\frac{1.700\ (300)}{340{,}000} = 1.5 \text{ lb. air per cycle}$$

With 550 psi available pressure drop in the receiver the volume necessary to deliver 1.5 pounds of air is

$$1.5 = \frac{550\, v}{340{,}000}$$

$$v = 930 \text{ cu. in.}$$

When numerous repeated cycles of operation are required, a more rigorous analysis is necessary. The basic gas laws yield the following formula, except the exponent has been modified by experimental data:

$$P_2 = P_1 \left(\frac{V_1}{V_1 + V_2}\right)^{1.1N} \qquad (8)$$

where N = number of repeated cycles.

As another example, it is desired to determine the receiver volume at 3000 psi working pressure that must store sufficient air to operate a cylinder of 30

cubic inches displacement per cycle seven times at 1500 psi minimum before recharge. Assuming no leakage and no pressure regulators in the system

$$1500 = 3000 \left(\frac{v_1}{v_1 + 30} \right)^{1.1 \times 7}$$

$$v_1 = 320 \text{ cu. ins.}$$

The second method of analysis accounts for the reheating of the air in the receiver during the multiple cycles of operation. This energy comes from the heat capacity of the receiver walls and the surrounding atmosphere. The exponent 1.1 is taken from actual practice to allow for this and other minor factors.

On weight consideration only, *Fig.* 8 indicates that there is not too much to be gained by going to increasingly higher pressures. The increased weight of the compressor and associated equipment when taken with the receiver and end use equipment indicate the optimum operating pressure for minimum weight to be below 1500 psi. However, in aircraft, space requirements dictate current working pressures of 3000 psi and consideration is being given to 6000 psi systems for this same reason.

As indicated in *Fig.* 1, it is undesirable to allow receiver temperature to fall appreciably after charging. It is even more undesirable to take a very rapid pressure depletion over a large ratio (final to initial pressure). Expansion conditions within the vessel approach adiabatic (1.4) and a very large temperature drop occurs leaving cold dense air in the receiver. Heating of this air may be desirable for special applications such as gas turbine or diesel engine starting. If the outflowing air is heated by combustion of fuel in it or by heat exchangers, some of this warm air may be recirculated to the receiver so that less weight of gas is left in the receiver at the end of the operation.

Even for commercial installation where weight is not a prime consideration, it is desirable to make the receiver no thicker and heavier than necessary for economical manufacture. In order to have a common measure for all types of receivers the author has used a somewhat arbitrary definition for some years and found it useful.

$$I = \frac{BV}{W}$$

where I = pressure vessel index; B = burst pressure, psi; V = volume, cu in., and W = weight, lb. In *Fig.* 9 is a chart indicating what can be expected from current practice.

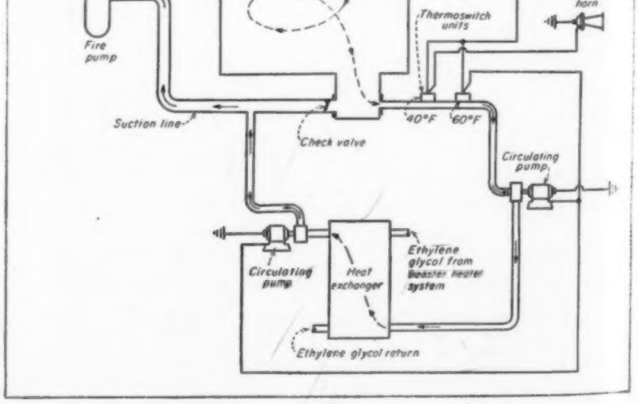
Although the designer is faced with supplying the lightest, the smallest, or the cheapest possible receiver for a given installation, he can never forget that safety is really the prime consideration—all others are secondary. To alter and quote an old proverb, "High pressure air, like the sea, is horribly unforgiving of carelessness." Never let the designer be accused of carelessness.

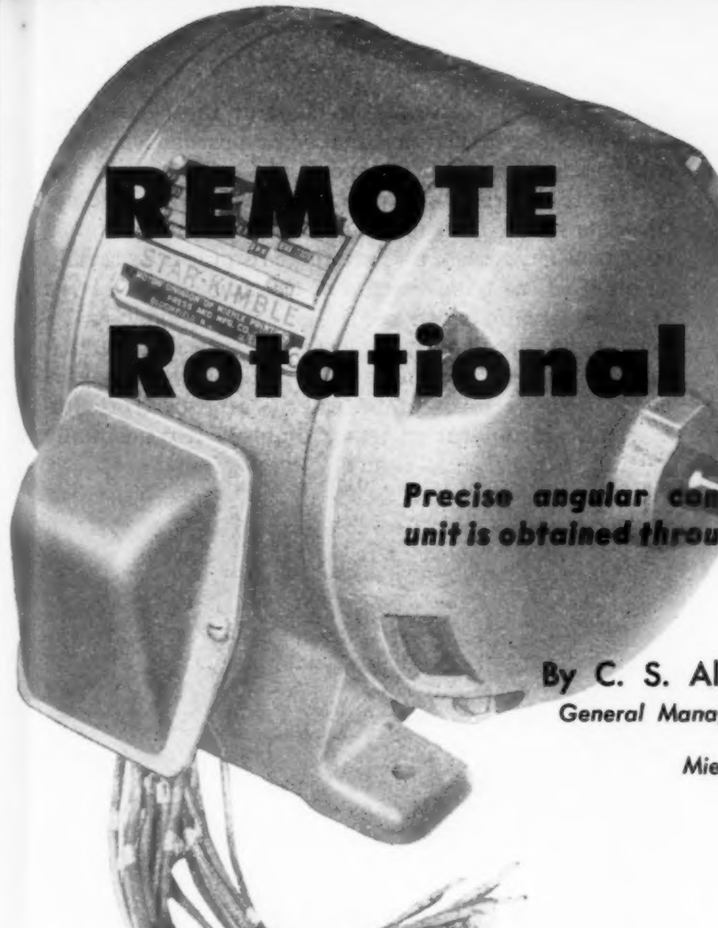
Circulating System Prevents Fire Truck Freeze-Ups

AN AUTOMATICALLY actuated circulation system, controlled by thermostats, keeps the water in fire trucks from freezing despite exposure to temperatures as low as −65 F. This circulation system, installed in the new Type 0-10 fire trucks designed for the U. S. Air Force by American-LaFrance-Foamite Corp., continuously circulates the fire-fighting water through a heat exchanger when the water temperature falls below a safe level.

Two circulation pumps draw the cold water from the 500-gallon water tank, pump it through a heat exchanger, where it is warmed, and return it through the fire pump piping to the tank. Heat is supplied to the heat exchanger by a circulating liquid, usually ethylene glycol, which is brought to temperature by a thermostatically-controlled 90,000 BTU gasoline heater.

The circulating pumps are controlled by a precision thermostat manufactured by Fenwal Inc. The thermostat, set at 60 F, is installed in an exposed section of the water line to assure earliest possible contact with low ambient temperatures. To warn personnel that some malfunction has permitted the water temperature to fall to a dangerously low level, a second thermostat is installed in the circulating line. This thermostat, whose contacts are adjusted to close at 40 F, is wired to the vehicle's horn. This warning notifies operating personnel of a malfunction so that they can make repairs before freezing damage occurs.





REMOTE Rotational Positioning

Precise angular control with this dc unit is obtained through simple switches

By **C. S. Allen** and **Harris Shapiro**

General Manager — *Chief Electrical Engineer*

Star-Kimble Motor Div.
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MANY types of positioners have been developed for the remote control of levers, dials, valves, and other adjusters on various types of equipment. In principle and operation, some of these are mechanical, some are hydraulic, and still others are electric. At times a combination of two or even all three of these approaches has been used. Each positioning device has particular characteristics which make it suitable for specific applications.

A new positioning device, known as a dc positioning motor, has been developed by Star-Kimble. This simple rotary, electromagnetic unit indexes to a number of predetermined positions. It can be remotely energized and directionally controlled by a rotary selector switch, push-button control or other contact-making mechanism. Several of these positioners can be powered from the same master switch allowing the remote start and control of a number of operations simultaneously. For its size this positioning device develops a high torque which provides a definite and fast response in following any control switch.

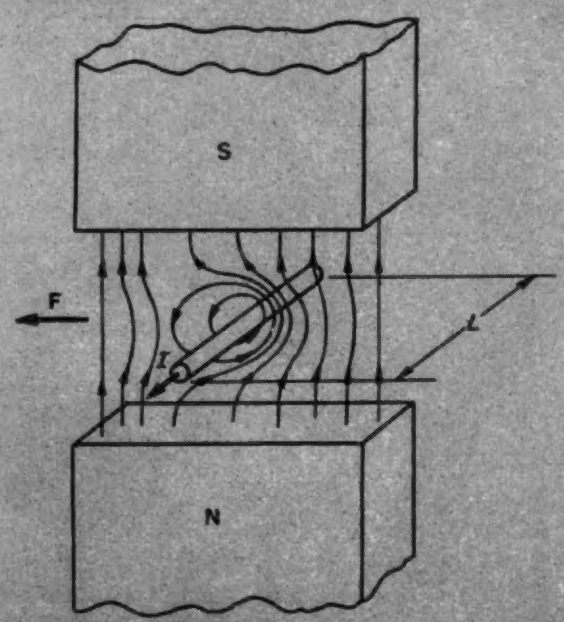
Theory of Operation: The principle of operation of the positioning motor is perhaps best explained by Ampere's fundamental law of magnetism. It states that a current-carrying conductor located in a magnetic field and at right angles to the field will be pushed by a force proportional to the flux density of the field, the current flowing through the conductor, and the length of the conductor that is actively in the field. The attraction and repulsion effects of the conductor field and the magnet field are shown in *Fig.* 1 and are expressed mathematically by

$$F = BIL$$

where F = force, B = flux density, I = conductor current, and L = active conductor length.

Consider a stator which consists of conductors arranged in slots of an iron cylinder. Current-carrying conductors in an iron core are said to produce an *electromagnetic* field. The forces created by this field can be examined with the law of magnetism in mind.

Fig. 1—Force on a current-carrying conductor located in magnetic field and at right angles to field is proportional to field flux density and conductor current and length



Let current flow in the same direction through half of the conductors on one side of the stator and equal current flow in the opposite direction through the other half. A bar magnet, free to rotate, is placed at right angles in this stator with its axis at the center. It will align or position itself as depicted in *Fig.* 2 because of a balance of forces of the stator field and the bar-magnet field. This concept is the basis of the dc positioner which is simply a rotor in the form of a magnet that aligns itself in the electromagnetic field of a stator. Orientation changes of this stator field provide a number of accurately controlled rotor positions.

Design: The positioner, shown disassembled in *Fig.* 3, is fundamentally a dc motor of special construction. The rotor in its simplest and most common form is a two-pole, permanent magnet with each pole having a soft-iron pole-shoe. For special application the torque of the positioning motor can be increased by use of a wound dc rotor with slip rings in place of the permanent-magnet rotor. This feature would be especially valuable in intermittent duty units to produce a high torque without the danger of demagnetizing a permanent-magnet rotor.

The stator is high-permeability iron with slots which contain a continuous full-pitch, double-layer lap winding. Leads are brought out from each connected pair of coils which make up this stator winding. There will be the same number of leads as slots and there may be as many slots as it is practical to have in a given stator diameter. It becomes apparent that a greater number of slots can be more easily incorporated as the stator diameter is increased.

Positioning Control: Direct current is used for positioning power and is usually applied to only two diametrically opposite stator leads at the same time. Generally the maximum number of positions possible will equal the number of leads. Since the transmitter or remote-control mechanism for operating the dc positioning motor is usually nothing more than a simple multiposition switch, its particular form can be readily adapted to the requirements of the application. If all available positions are needed, a simple rotary switch with a pointer indicating each position will be adequate. The switch can be built in the form of a keyboard with each key representing a desired orientation. If an application requires only a few unequally

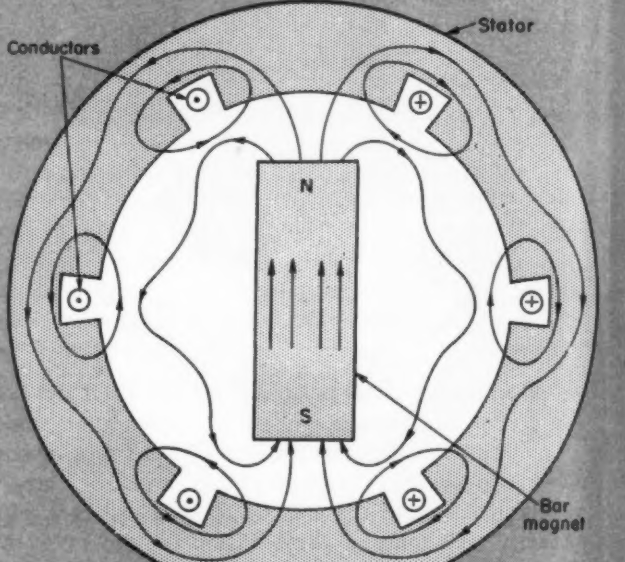


Fig. 2—Above—Magnetic forces caused by equal current flow in all conductors in the directions indicated will position magnet as shown. Conductors with dots represent current flowing out of paper; with crosses, flowing in

Fig. 3—Left—Positioning motor showing frame, stator core, two-pole permanent magnet rotor and end bracket

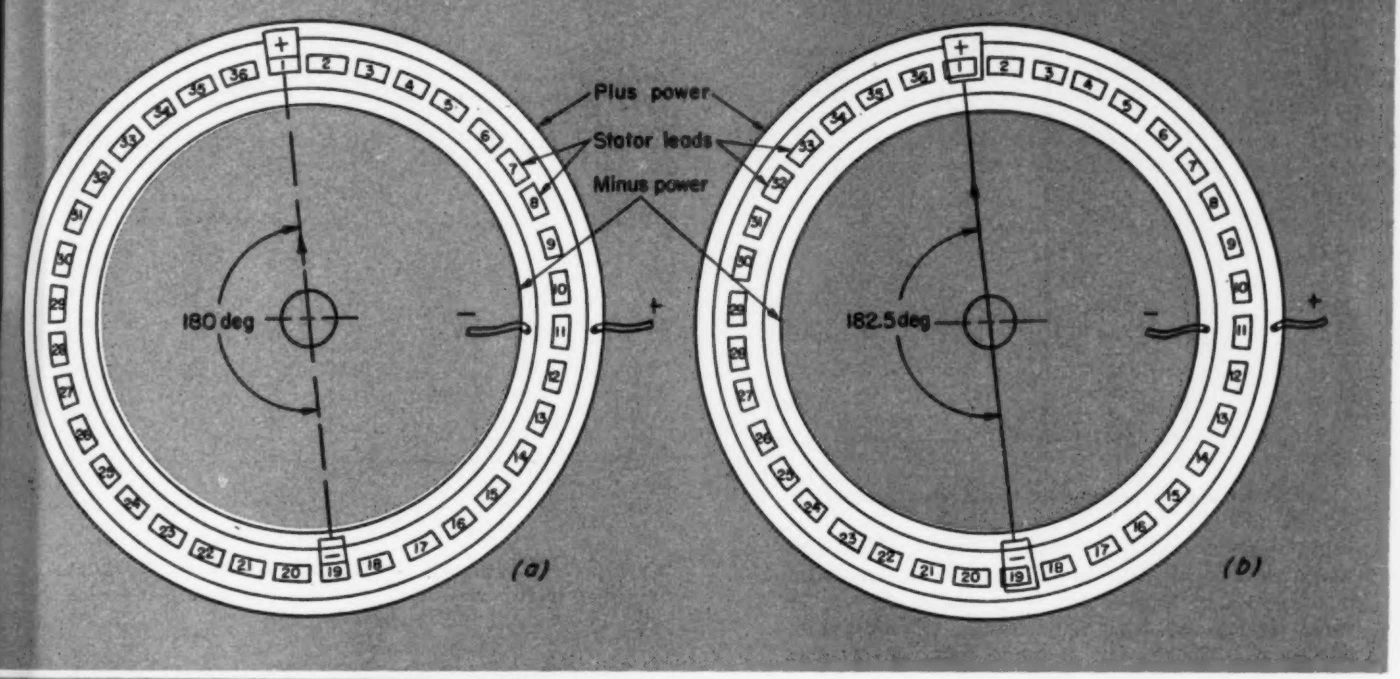
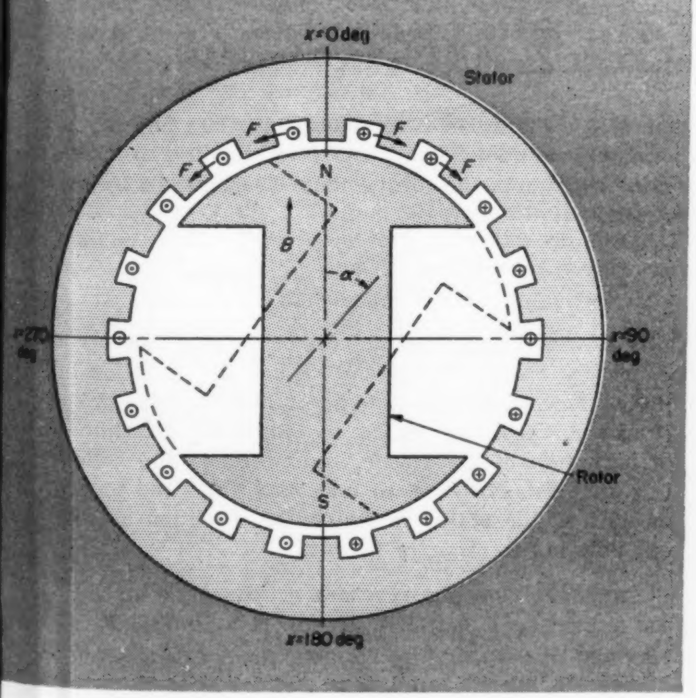


Fig. 4—Above—Rotary switch for obtaining 36 positions with a 36-lead motor, *a*, and rotary switch arrangement for obtaining 144 positions with a 36-lead motor, *b*

Fig. 5—Below—Cross-section of the motor



spaced positions, then only the necessary leads corresponding to these positions need be brought to the transmitter and the positioner can be operated by pushbuttons and relays.

A special switching arrangement makes it possible to increase the number of positions available by four times the number of stator slots or leads. For a 36-lead dc positioner, *Fig.* 4*a* schematically shows a multiposition switch for applying power to two diametrically opposite leads. When plus power is supplied to lead 1, minus power is supplied to lead 19; when plus power is supplied to lead 2, minus power is supplied to lead 20, etc. Thus 36 definite positions are available from this switch.

In *Fig.* 4*b* the same switch is shown except the positive power contact is displaced clockwise 2.5 degrees while the negative power contact is not moved with respect to the switch shaft. As this switch is initially turned from leads 1 and 19 toward leads 2 and 20, respectively, the positive contact first bridges leads 1 and 2 while lead 19 alone receives minus power. This connection rotates the orientation of the stator field 2.5 degrees and thus the rotor turns 2.5 degrees to re-establish equilibrium. If the switch is turned further, leads 19 and 20 are bridged while 1 and 2 are still bridged. This condition provides another 2.5-degree change in the stator field followed by a corresponding adjustment of rotor direction. Further advance of the switch causes the positive contact to leave lead 1 and supplies plus power to lead 2 only while leads 19 and 20 continue to receive minus power. This contact arrangement rotates the rotor 2.5 degrees more. Finally, the negative contact leaves lead 19 and power is applied to leads 2 and 20 only. Another 2.5-degree rotor change results. Therefore, this special switch provides for the accurate orientation of the rotor of this 36-lead motor in 144 positions or every 2.5 degrees.

Positioning Torque: When a dc voltage is applied to two diametrically opposite leads of the positioning motor, *Fig.* 5, half of the stator winding will carry current in one direction and half in the other direction, as shown graphically in *Fig.* 6 by the dotted line, N_sI. The strength of the magnetic field is assumed

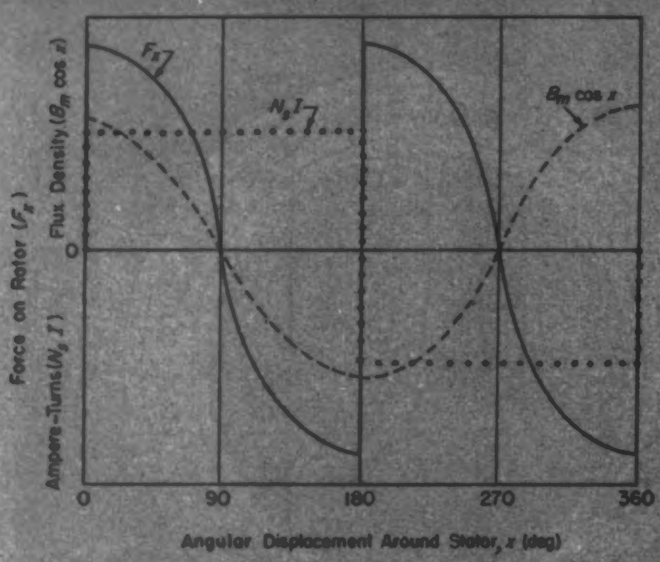


Fig. 6—Distribution of stator current, N_sI, and magnetic flux density in air gap, B_m cos x, from 0 to 360 deg with rotor in position shown by solid outline in *Fig. 5*. Product of these two curves is proportional to force and thus torque

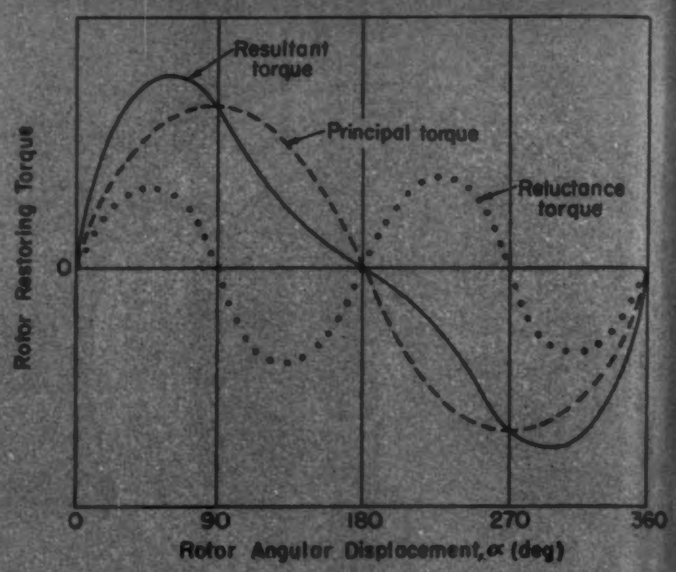


Fig. 7—Restoring torque of permanent magnet rotor is the resultant of principal and reluctance torque curves

to be sinusoidally distributed in the air-gap as shown in *Fig.* 6 by the dashed curve, B_m cos x. This condition closely approximates reality.

The forces at any point in the air-gap exerted on the rotor, with it in the position indicated by the solid outline in *Fig.* 5, are proportional to the products of the two curves N_sI and $B_m \cos x$. This product is shown by the solid line in *Fig.* 6. For this rotor position, the average force (and therefore torque) is zero, which is to be expected since the stator and rotor fields are in line or in equilibrium.

When the rotor is turned from its equilibrium position as shown by the dashed outline in *Fig.* 5 by an angle α, or equivalently, if a dc voltage is applied to a different pair of diametrically opposite stator leads removed from the original by the angle α, a restoring torque is created. The magnitude of this torque is the sum of the forces around the rotor times the radius of the rotor, or symbolically

$$T_a = \sum_{x=0}^{x=2\pi} F_x \left(\frac{D}{2}\right)$$

where T_a = restoring torque at angle α, α = rotor angular displacement from point of zero torque, F_x = force on rotor at any point x, and D = rotor diameter. But since

$$F = B\,I\,L$$

the general equation can be written

$$T_a = \frac{1}{2\pi} \sum_{x=0}^{x=2\pi} (B\,I\,L)_x\,(\Delta x) \left(\frac{D}{2}\right)$$

More specifically

$$T_a = \frac{1}{2\pi} \sum_{x=0}^{x=2\pi} [L\,N_s\,I\,B_m \cos(x+\alpha)]\,(\Delta x) \left(\frac{D}{2}\right)$$

where N_s = number of conductors per slot and B_m = maximum magnetic flux density in the air-gap. Combining constants, the torque could be evaluated by the integral

$$\frac{DLN_sIB_m}{4\pi} \int_0^{2\pi} \cos(x+\alpha)\,dx$$

if the flow of the current, *I*, were in the same direction completely around the stator. Since the direction of current flow with respect to the stator changes every π radians, the restoring torque could be determined for each half of the stator circumference and added. It is simplest, however, to take two times the torque for half of the stator in which the direction of current flow is the same, or

$$T_a = \frac{DLN_sIB_m}{2\pi} \int_0^{\pi} \cos(x+\alpha)\,dx$$

Integrating gives

$$T_a = K\,N_s\,I\,B_m \sin\alpha$$

It should be noted that the foregoing derivation assumes that the conductors themselves lie in the magnetic field and all force is exerted on them. Actually this is not the case since the conductors are embedded in the stator slots surrounded by iron of high permeability which carry most of the flux. Therefore the forces are redistributed and most of the thrust is on

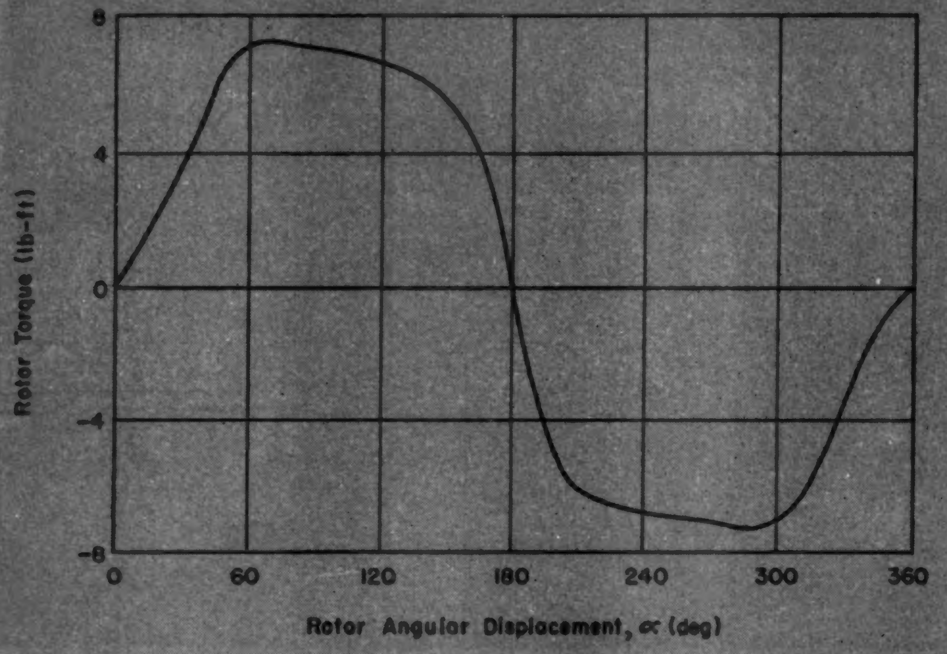


Fig. 8—Test data of a dc positioner with a permanent magnet rotor when rated dc power was applied to two diametrically opposite leads while the shaft was rotated in 10-deg steps from 0 to 360 deg

the stator core and little is on the conductors. The total force, however, remains the same and the torque equation is valid.

The torque-angle characteristic of a positioning motor with a wound rotor will approximate a sine wave times a constant as already discussed. This is shown by the dashed curve in *Fig.* 7 and is referred to as the principal torque.

When a permanent-magnet rotor is used, a smaller secondary torque component is introduced in addition to the principal component of torque produced by the interaction of the stator current and the permanent-magnet field. This secondary torque is produced because the air-gap is larger in the interpolar space than it is along the polar axis. Known as a reluctance torque since it is caused by the difference in reluctance between the interpolar and polar axes, it always tends to return the rotor to the position which will make the flux produced by the stator field a maximum. *Fig.* 5 shows that its value must be zero when $\alpha = 0$ or $\pi/2$ radians. This "second harmonic torque" as shown by the dotted curve in *Fig.* 7 distorts the resultant torque curve. The resultant curve, which is the algebraic sum of the principal and reluctance components of torque, is shown by the solid curve in *Fig.* 7. The reluctance torque does not appear in a positioning motor with a wound rotor because it has a uniform air-gap over the entire rotor circumference.

Test data from an actual permanent-magnet positioner similar to the one shown in *Fig.* 3 are plotted in Fig. 8. Approximately 10 inches long and 8 inches in diameter, the 36-slot unit was designed to operate on 28 volts dc. The current was 4.8 amperes and thus the power consumed was 135 watts. For test purposes the rotor was held stationary with an arm connected to a spring balance for measuring the torque and the power was successively applied to adjacent leads. The torques indicated can also be considered as those that would be developed if the dc supply power remained connected to two diametrically opposite taps and the shaft were rotated in 10-degree steps from 0 to 360 degrees.

The maximum torque was about 7 lb-ft. A substantially high torque was reached by 45 degrees and remained more or less constant to about 135 degrees and then dropped back to zero. The other half of the curve is similar but the torque is in the reverse direction. It is apparent that this is a desirable torque-angle characteristic, because slight rotational displacements of the rotor cause sharp increases in torque and thereby produce accurate positioning.

Sizes and Applications: The permanent-magnet rotor provides for a simple and rugged rotor construction. If properly stabilized, it should be completely satisfactory. However, should operating conditions require it, a wound rotor can be substituted. Since the stator is essentially a standard motor stator, it is apparent that large sizes can be built. The torque ratings of motors in the larger frames can be further increased by the use of Class B and Class H insulation or through the use of external motor-driven blowers of sufficient size to dissipate the losses and to maintain safe temperature limits. The dc positioning motor can be built in practically any size.

Typical applications of this unit are positioning hydraulic valves, adjusting heat controls for railroad cars, turning tuning dials for aircraft radios, positioning speed control rheostats, controlling variable-speed pulleys, and spot location of automatic welders. Besides the rotational and positioning uses, it shows promise for supplying a predetermined pressure or torque, for measuring pressure or torque, and for indicating position or magnitude of a remote operation.

Selection and Application of ROLLER BEARINGS

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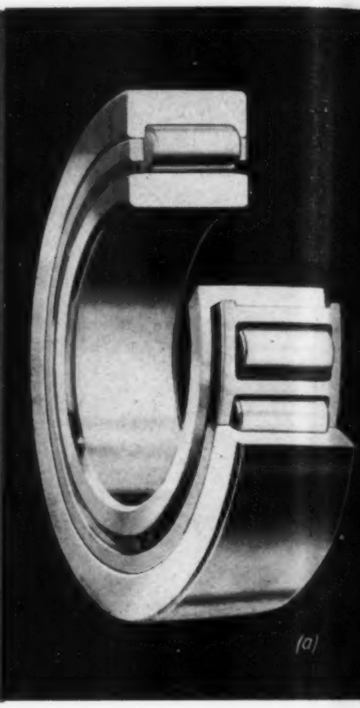

(a)

SERVICE performance of any machine using roller bearings depends on the designer's ability to visualize the nature and magnitude of service loads, and to select bearings to withstand these loads. The bearing must have performance characteristics suited to expected load and alignment conditions; bearing mounting must be designed to assure proper load distribution; and bearing fitting practices must suit the temperature and load conditions prevailing in service.

The selection problem can be reduced to two fundamentals. First, the nature of the supporting housing must be defined; then the type of bearing must be selected. Choice of type of bearing depends on nature of the motion, rigidity of the installation, and directions of loading. Selection of bearing size involves consideration of speed, magnitude and nature of the load, and desired life expectancy. If probable service loads are indefinite and shock loads are likely, an application factor based on experience should be provided to allow for these contingencies.

Choosing a Bearing Type: Within rather broad limits, the type of bearing selected for a given application is a matter of design choice; all types have been used in nearly all applications. This freedom is usually achieved by design of the structure used to support and house the bearing. The designer has three basic types of roller bearings to choose from—straight, tapered and self-aligning—and each type of bearing has its individual characteristics. Typical bearings of each type are shown in *Fig.* 1.

Basically, the straight roller bearing is suited to resist radial loads in installations where shaft and housing deflection do not disrupt the normal even load distribution over the face of the bearing. Conventional straight roller bearings will not resist thrust or axial loads; however, they can be obtained with guide lips on both the inner and outer races as shown in *Fig.* 2. Constructed in this manner, they resist the light loads required to maintain axial shaft location in installations where the shaft is subjected to very minor thrust, such as that occurring with belt or chain drives. Heavy thrust loads, however, cannot be carried.

Tapered and self-aligning roller bearings are designed to carry heavy radial and thrust loads. The self-aligning bearing can be used in installations where shaft or housing deflections are excessive, and where unavoidable manufacturing variations create adverse alignment between the shaft and housing. Straight and tapered roller bearings can also be applied to the latter type of installation if they are mounted in self-aligning housings.

If straight or tapered roller bearings are applied where excessive deflection exists between the shaft and housing, life of the bearing will be reduced materially, because the normal load distribution across the face of the roller is disturbed. In an ideally loaded straight roller bearing, the load is uniformly distributed over the length of the roller. This uniform load distribution produces a roller stress pattern as shown in *Fig.* 3*a*. Unit stress is not uniform over the length of the roller, and the highest stresses occur at ends of the

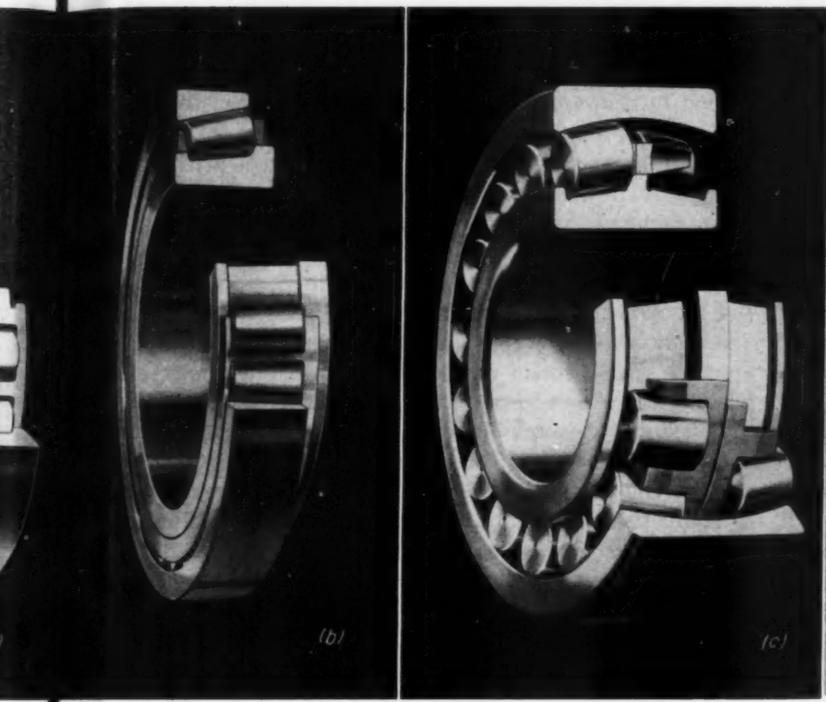
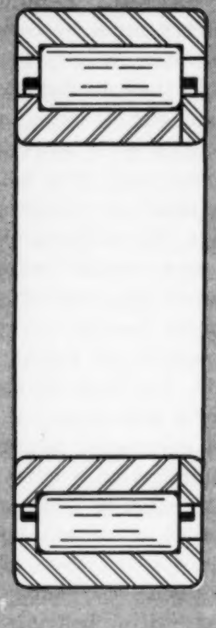


Fig. 1—Left — Basic types of roller bearings are: *a*—straight, for radial loads; *b*—tapered, for radial and thrust loads; *c*—self-aligning, for radial and thrust loads and to accommodate misalignment

Fig. 2 — Right — Straight roller bearings with guide lips on inner and outer races can be used for light thrust loads



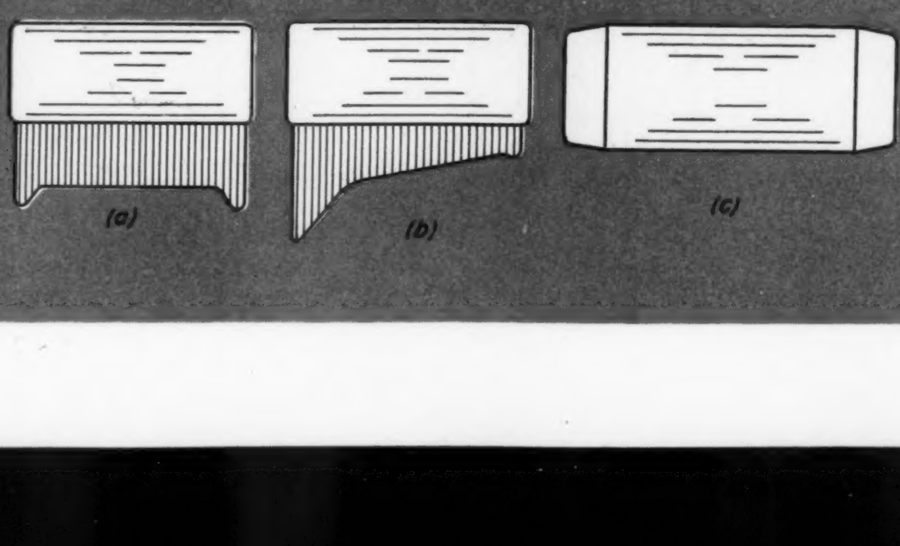
roller. If this roller is subjected to a nonuniform load due to excessive shaft deflection, one end of the roller must support an increased load, while the load on the other end of the roller is decreased. This load condition produces a stress pattern as shown in *Fig. 3b*. Stress at one end of the roller is much higher than at the opposite end and, therefore, the load capacity of the bearing for a given life is materially decreased.

Sensitivity of straight roller bearings to shaft and housing deflection can be decreased by relieving the ends of the rollers, *Fig. 3c*. When roller end-relief is properly proportioned, stress across the face of the roller is uniform, and the high stress points at the end of the roller are eliminated. End-relief increases the useful load capacity of the bearing because it decreases the peak roller stress, and at the same time decreases sensitivity of the bearing to minor misalignment. However, it will not make the bearing suitable for major housing or shaft misalignment.

Designing the Housing: Since load capacities of both straight and tapered roller bearings are affected by misalignment, combined housing and shaft deflections must be limited to assure good bearing performance. Experience has shown that total effective deflection at the bearing should be limited to approximately 0.0005-inch per inch of bearing width to assure good performance. As this deflection limit involves both the shaft and the bearing supporting housing, relative location of the bearing housing with reference to its support should be given careful consideration. Caution is particularly necessary in welded housings because of the simplicity of manufacturing housings off-center with respect to the supporting frame.

A typical off-center bearing housing in a welded assembly is shown in *Fig. 4a*. In this illustration, the

Fig. 3—Normal stress pattern for a straight roller bearing, *a*, shows a sharp increase in stress at one end, *b*, if shaft deflection is excessive. Properly proportioned end-relief on the rollers, *c*, eliminates the high stress points at ends of the rollers shown in *a*



center of the bearing is not coincident with the center of the case wall. Bending moment resulting from this asymmetry must be resisted by the case wall since it may produce deflections which limit bearing performance. If the bearing does not carry thrust loads, this deflection can be substantially eliminated by mounting the bearing with its center coincident with the center of the case wall, *Fig. 4b*.

If a shaft is carried on single row tapered roller bearings, the housing must be designed to resist both radial and thrust loads. If the bearing is subjected to a pure radial load, this radial load produces a thrust component because of the taper. If the bearing carries a combined radial and thrust load, the housing must be designed to resist each of these loads with minimum deflection. If the bearings are carried in a welded structure, minimum deflection can usually be accomplished with ribs, *Fig. 5*.

If the housing is not sufficiently rigid to resist these loads with minimum deflection, adverse load distribution across the face of the bearing will reduce the bearing life. Excessive housing deflection also results in poor initial bearing adjustment and produces effects identical with those resulting from worn or loosely adjusted bearings. Looseness from this condition may cause a shift in the location of the shaft, which may adversely affect the operation of machine elements driving the shaft. If the driving elements are gears, this alignment shift may create an unfavorable load distribution across their working faces.

Specifying Mounting Tolerances: If they are assembled with zero diametral clearance and subjected to minor misalignment, straight roller bearings will create extremely high thrust loads. Sensitivity to misalignment decreases as the diametral clearance is increased. Within limits, the diametral clearance is controlled by the amount of interference between the inner and outer races and their supporting surfaces. These interferences should be in accordance with the bearing manufacturer's recommendations.

An unusual example of a straight roller bearing application which gave unsatisfactory service due to the thrust loads created by the bearing in the connecting rod assembly is shown in *Fig. 6*. This connecting rod assembly was used in a high-pressure power pump, and the thrust loads developed in the connecting rod bearing caused excessive crosshead wear. In the original design, the connecting rod was located laterally at the crosshead end, and was allowed to float laterally on the straight roller bearing on the crankthrow to compensate for manufacturing variations. When it was realized that crosshead failures were caused by the thrust loads developed in the straight roller bearings, the design of the crankthrow bearing was changed to provide lateral retention. In the new design the crankthrow bearing was changed to the design shown in *Fig. 2*, thus retaining the connecting rod laterally at the crankshaft. Lateral clearance between the small end of the connecting rod and the crosshead was increased to provide lateral movement at this point to allow for manufacturing variations. This change relieved the crosshead of the lateral loads and eliminated failures.

To assure freedom from bearing race creep or rotation on the shaft seat or in the housing, bearing mounting tolerances must suit the directional nature of the load. For example, if the outer race of a bear-

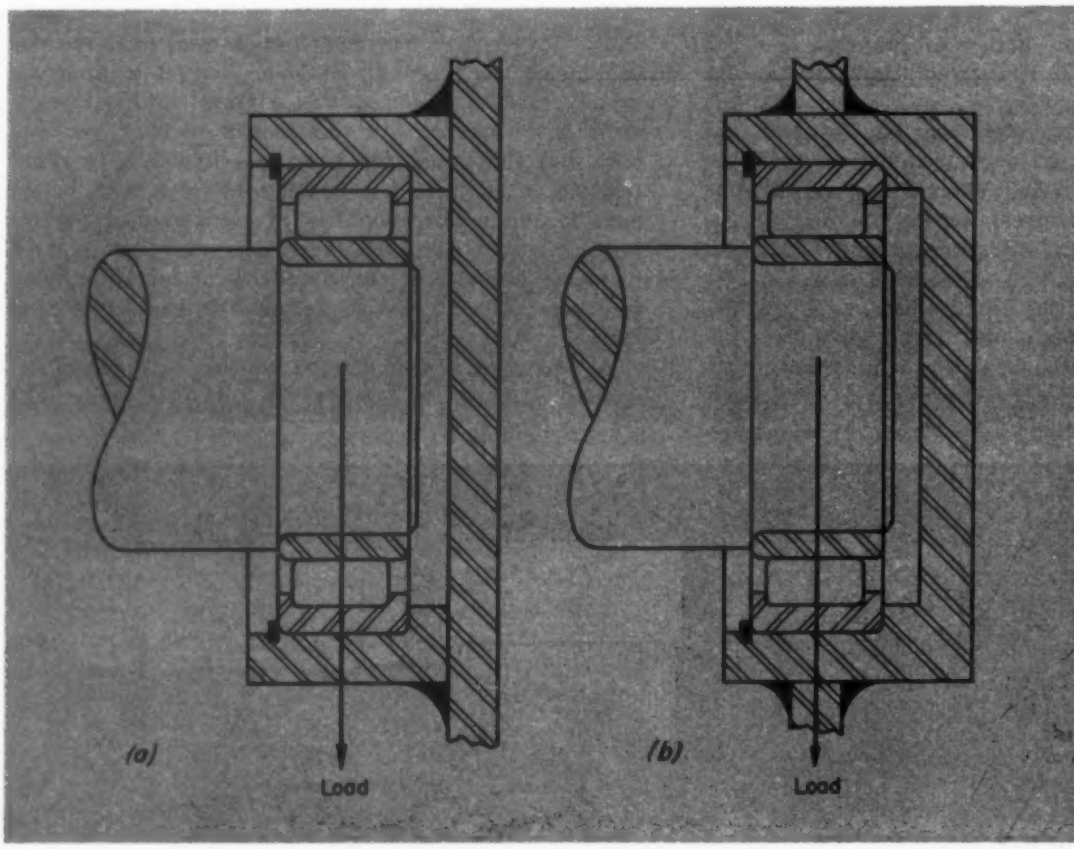


Fig. 4 — Undesirable deflection may result if the bearing load is not adequately supported, *a*, but can be eliminated by proper location of the bearing housing, *b*

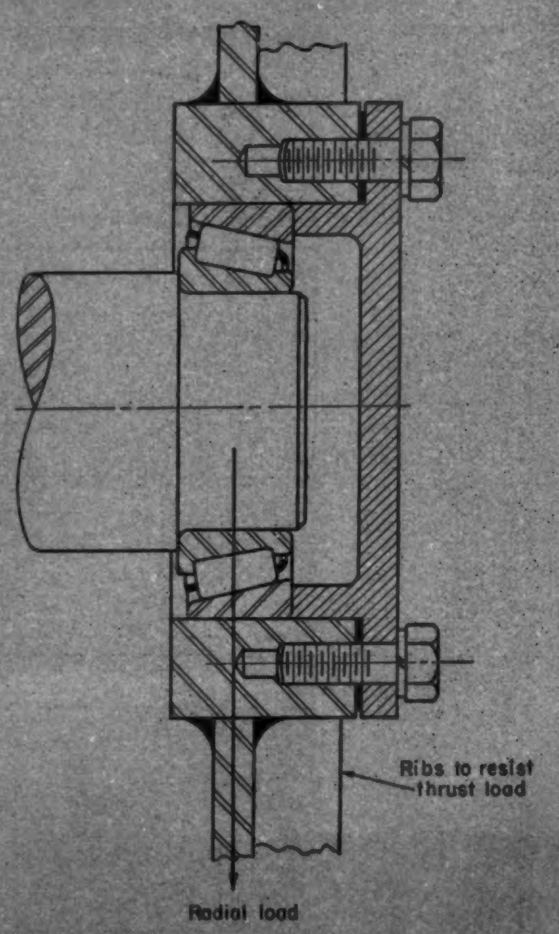


Fig. 5—Ribs may be used with tapered roller bearings to resist normal thrust loads

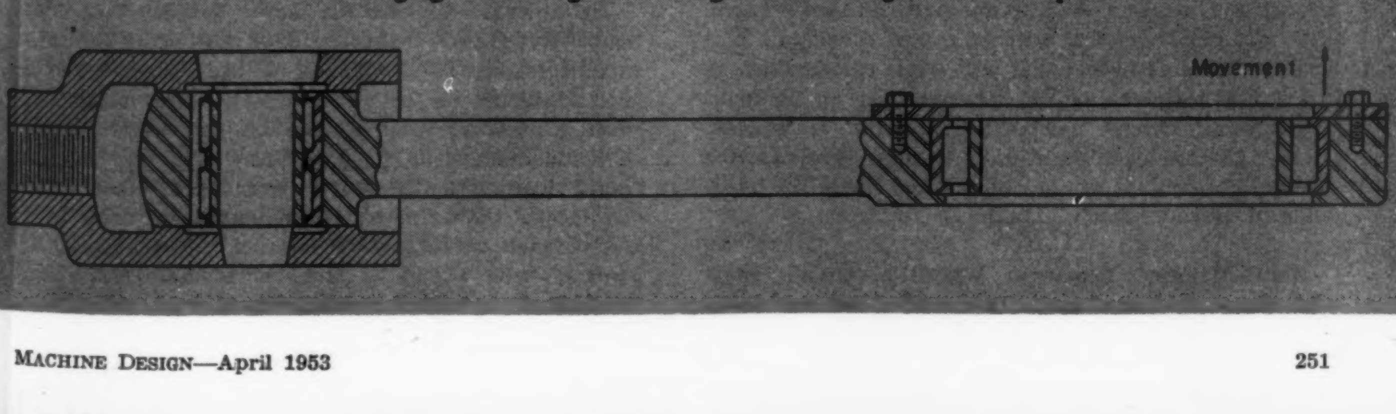
ing is not an interference fit in its housing, and the direction of the load rotates with relation to a fixed point on the housing, the bearing race creeps or rotates in the housing. Relative motion thus produced between the bearing race and housing causes housing wear, resulting in excessive looseness between these two parts. In this case, the relative motion between the two elements is the result of the bearing race acting as a planet gear in an epicyclic gear train. If the load revolves with reference to a fixed point on the bearing housing, *Fig.* 7, the point of contact between the outer race and its bore also revolves. Since circumferences of the two elements are unequal, relative motion exists between the two parts. This same action occurs on the inner race if it is not an interference fit on the shaft and the load direction revolves with reference to a fixed point on the shaft. In most designs, load direction is constant with respect to a fixed point on either the shaft or housing. Therefore, standard roller bearings are designed to have one race mounted with an interference fit and the other mounted with approximately zero tolerance.

If unitized bearings, such as straight and self-aligning roller bearings, are applied with an interference fit between both races and their supporting surfaces, special bearings with extra amounts of internal clearance should be used. If standard bearings are used, race expansion and contraction resulting from the interference fit may cause the bearing to be preloaded, thus reducing the useful load capacity of the bearing. In a straight roller bearing, this preload may cause the development of excessive end-thrust loads resulting from the tendency toward axial movement created by minor dimensional variations in the bearing.

Bearing Adjustment Methods: If the shaft is carried on adapter-mounted bearings similar to that shown in *Fig.* 8, internal clearance is controlled by the tightness of the adapter drawnut, and the adjustment should be controlled by the internal clearance in the bearing. This control can be accomplished by measuring the end movement of the shaft relative to the housing, or by measuring the internal clearance in the bearing with a feeler gage. These clearances must be in accordance with the bearing manufacturer's recommendations if satisfactory bearing service is to be achieved.

The problem of bearing adjustment must be considered during selection of the mounting arrangement for tapered roller bearings. If the bearing races are to be mounted directly in the housing and on the shaft, it is desirable to make the bearing adjustment by moving the member which is a slip fit on its supporting surface. In the more common applications where the shaft rotates and the outer race is a tap fit in the

Fig. 6—Lateral movement of the connecting rod in this design caused excessive crosshead wear. Changing the bearing to the design shown in *Fig.* 2 solved the problem



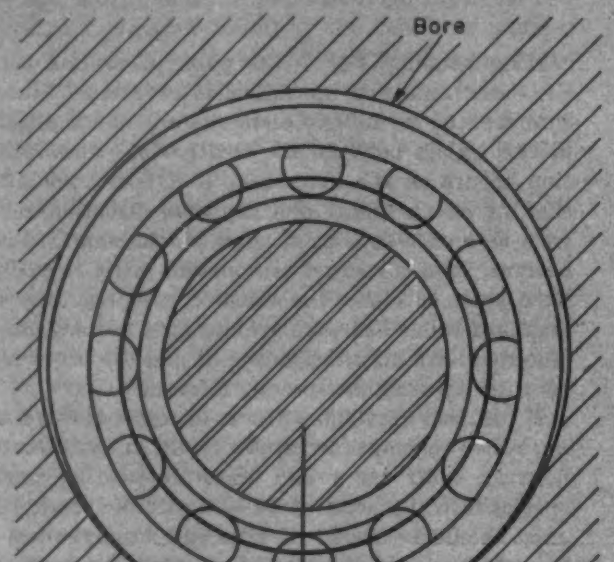
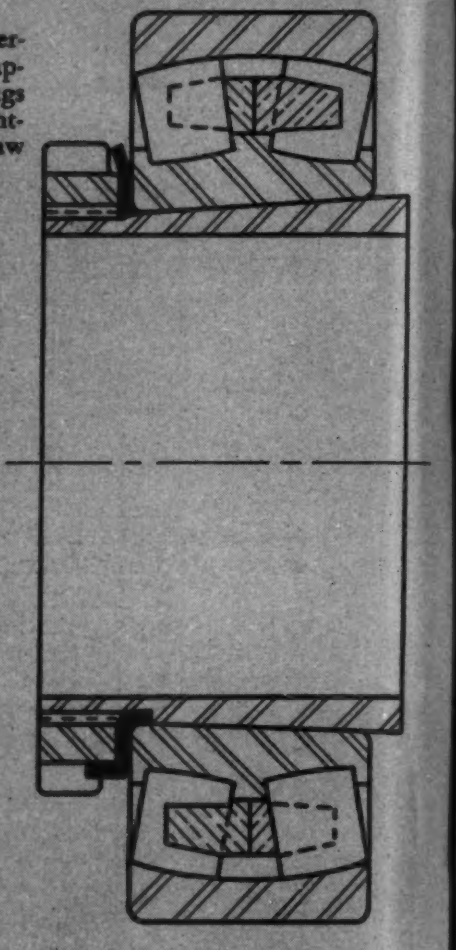


Fig. 7—If the bearing is not an interference fit in its housing and direction of the load rotates, creep or rotation of the bearing produces excessive housing wear

Fig. 8—Right—Internal clearance on adapter-mounted bearings is controlled by tightening the adapter draw nut



housing, bearing adjustment should be attained by moving the outer race. This can be accomplished with shims, *Fig. 9a*, or with a threaded member, *b*. If the load direction rotates relative to the housing and it is desirable to adjust the bearing by moving the outer race, the outer race should be indirectly mounted, *Fig. 9c*. In this mounting, a press fit between the outer race and its bore is used to prevent race creep. Bearing adjustment is attained with shims between the flange of the carrier and the housing.

When tapered roller bearings are adjusted by moving the inner race on the shaft, the adjusting device must be designed to resist wear which will result in loss of adjustment. This wear usually occurs on the face of the adjusting nut or washer; it can be prevented by hardening the faces which contact the bearing race. Since the standard SAE-type bearing retainer nut and lockwasher does not have hardened faces, it should not be used to adjust tapered bearings. If it is used, the soft lockwasher will wear, causing loss of bearing adjustment, or the retaining key on the lockwasher may break, permitting the nut to become loose. Double locknuts, *Fig. 9d*, can be used. In this design, the contacting face of the nut should be hardened to prevent wear.

Shaft-Mounted Bearings: Where maximum shaft strength is desired in applications using straight roller bearings, the shaft is usually hardened to serve as an inner race for the bearing. The shaft should be made of an alloy grade steel suitable for roller bearing service, and it must be hardened to 60 Rockwell C minimum if the full load capacity of the bearing is to be achieved. If shaft hardness is less than 60 Rockwell C, the load capacity of the bearing will be reduced, as shown graphically in *Fig. 10*. Note that a small decrease in hardness produces a material reduction in load capacity. If the shaft is casehardened, the case must extend below the point of maximum subsurface stress in the shaft. Experience has shown that a case depth of 0.035-inch for a 1-inch shaft and 0.035-inch plus 0.005-inch for each inch of diameter over 1 is sufficient in most instances. To assure proper load distribution in the bearing, the shaft must be ground with a finish equivalent to that used on bearing parts.

In the event the installation is subjected to high temperature differentials, the effects of differential expansion must be considered. If heat is generated in or transmitted to the shaft, it causes the shaft and bearing inner race to expand. If the bearing outer race and housing do not expand at the same rate, internal clearance in the bearing is reduced. This action can subject the bearing to a preload which reduces its load capacity. In installations of this type, special bearings with increased amounts of diametral clearance should be used. If a shaft is carried on bearings

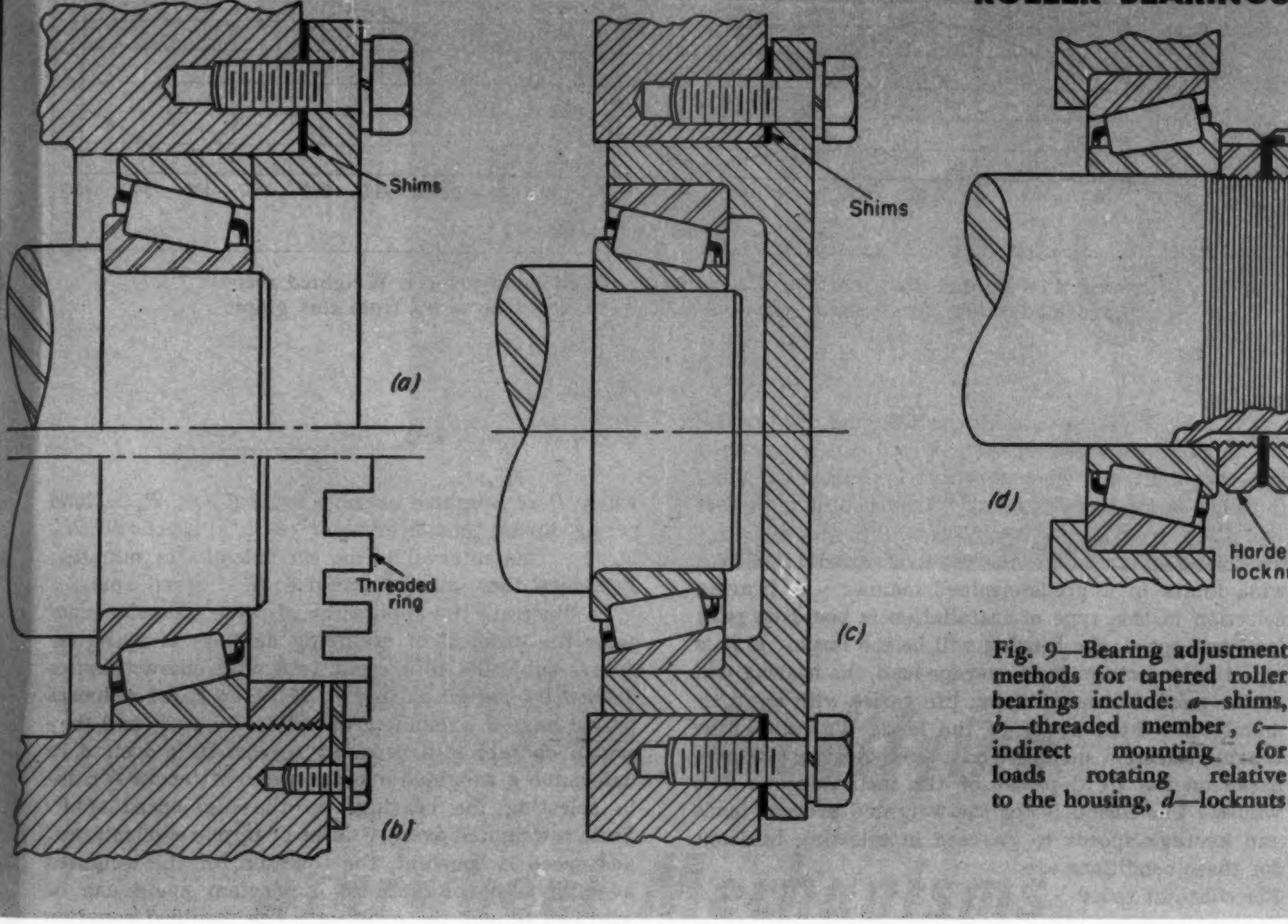


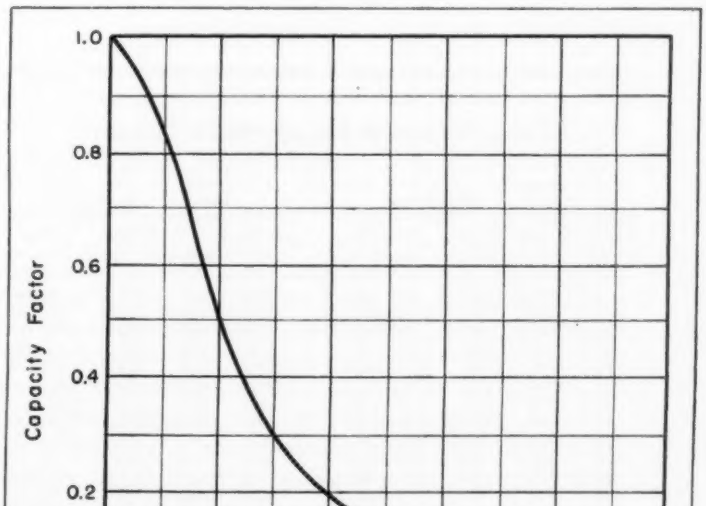
Fig. 9—Bearing adjustment methods for tapered roller bearings include: *a*—shims, *b*—threaded member, *c*—indirect mounting for loads rotating relative to the housing, *d*—locknuts

with thrust load capacity and both bearings are fixed axially on the shaft and in their housings, differential expansion between the shaft and housing can impose high thrust loads on the bearings. This condition can be eliminated by fixing the shaft at one bearing and permitting it to float laterally at the other.

Selecting the Proper Size: Size of the required bearing depends on the magnitude and nature of the load, speed, and the desired life expectancy. If a bearing failure might result in extensive damage to the machine or endanger human life, the bearing life expectancy should exceed the life expectancy of the other elements of the machine. When selecting life values, the fact that catalog life values are based on ideal conditions of alignment and lubrication must be considered. If these conditions are not ideal in the application, service life of the bearings will be reduced to a value less than that used in design calculations.

The most difficult factor to control and evaluate is lubrication. In low-speed applications the most common lubrication problems affecting bearing performance are insufficient lubricant and lubricants contaminated with corrosive or abrasive materials. In high-speed applications excessive lubrication can become an adverse factor in addition to those listed above. Faulty lubrication usually causes the bearing to wear by abrasion, and can materially reduce the life of the

Fig. 10—On bearings using the shaft as the inner race, load capacity decreases materially as shaft hardness drops off



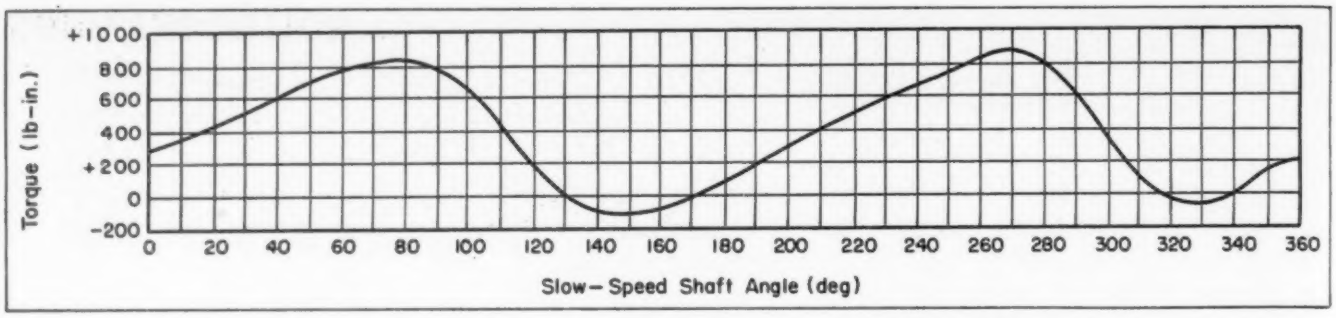


Fig. 11—Torque characteristics of a cyclically loaded speed reducer. Weighted average load values for bearing selection in Table 1 are determined from this graph

bearing.

Available literature covers the selection of bearings for applications where speed and load can be assumed to be constant, and when this data is properly applied, it yields satisfactory results. In general manufacturers' literature does not cover the selection of bearings for installations where the load or speed vary over wide limits in a predetermined manner. If bearing selection in this type of installation is based on peak load and speed, the bearing will be too large. If it is based on an arithmetical average load, the bearing will be too small because bearing life varies with approximately the third power of the load. Therefore, the bearing selection should be on a weighted average load, based on the third power of the instantaneous load values. The formulas for the weighted average loads and average speeds to be used in selecting bearings for these conditions are:

For constant speed

$$P = \sqrt[3]{\frac{P_1{}^3t_1 + P_2{}^3t_2 + \ldots + P_n{}^3t_n}{T}}$$

For variable speed

$$P = \sqrt[3]{\frac{P_1{}^3t_1N_1 + P_2{}^3t_2N_2 + \ldots + P_n{}^3t_nN_n}{t_1N_1 + t_2N_2 + \ldots + t_nN_n}}$$

Table 1—Speed-Reducer Shaft Torques

Shaft Angle (deg)	Slow Speed Shaft Torque (lb-in.)	High-Speed Shaft Torque (lb-in.)	High-Speed Shaft Torque³ (lb-in.)³	Shaft Angle (deg)	Slow Speed Shaft Torque (lb-in.)	High-Speed Shaft Torque (lb-in.)	High-Speed Shaft Torque³ (lb-in.)³
0	270	54	157,464	190	170	34	39,304
10	350	70	343,000	200	300	60	216,000
20	420	84	592,704	210	400	80	512,000
30	500	100	1,000,000	220	530	106	1,191,016
40	600	120	1,728,000	230	590	118	1,643,032
50	720	144	2,985,984	240	670	134	2,406,104
60	780	156	3,796,416	250	730	146	3,112,136
70	830	166	4,574,296	260	850	170	4,913,000
80	830	166	4,574,296	270	890	178	5,639,752
90	800	160	4,096,000	280	805	161	4,173,281
100	700	140	2,744,000	290	620	124	1,906,624
110	450	90	729,000	300	330	66	287,496
120	180	36	46,656	310	100	20	8000
130	0	0	0	320	−50	−10	−1000
140	−100	−20	−8000	330	−80	−16	−4096
150	−120	−24	−13,824	340	−10	−02	−8
160	− 80	−16	−4096	350	200	40	64,000
170	0	0	0				
180	90	18	5832		Total		53,454,369
					Mean cube		1,484,844

For average speed

$$N = \frac{t_1N_1 + t_2N_2 + \ldots + t_nN_n}{T}$$

where P = weighted average load; P_1, P_2, P_n = load acting during time intervals, t_1, t_2, t_n at speeds N_1, N_2, N_n; t = time interval during which load acts, minutes; T = total time interval, minutes; N = speed, rpm.

To illustrate the application of these formulas, consider the problem of designing a series of speed reducers subjected to a cyclic load with characteristics defined by the graph in *Fig.* 11. The speed reducers are to have a constant gear ratio of 5 to 1, and they are to operate at a constant speed. It is desired to determine a weighted average value of torque for use in selecting the bearings on the high-speed shaft. Since a weighted average value of torque is of interest, and speed is constant, the equation for the weighted average value of load for a constant speed can be modified to suit the problem. The modified equation is

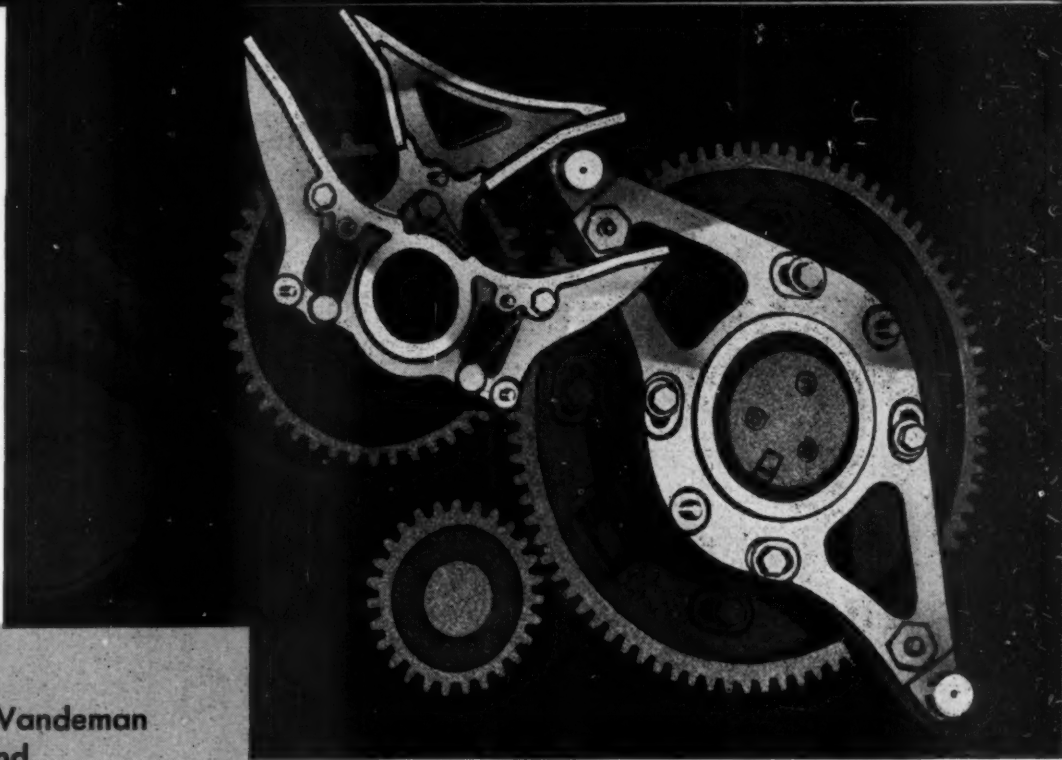
$$M = \sqrt[3]{m_1{}^3 + m_2{}^3 + \ldots + m_n{}^3}$$

where M = weighted average torque; and m_1, m_2, m_n = Average values of torque for small equal intervals of time.

Values of the slow-speed shaft torque as determined from the curve for 10-degree intervals, the high-speed shaft torque, and the cube of the high-speed shaft torque are calculated as shown in TABLE 1. Mean value of the cube of the torque for the high-speed shaft is 1,484,844, and the cube root of this figure is 114. The ratio of the weighted average torque to the peak torque is 0.642. This factor can be used to modify the peak torque loads when bearing loads are calculated, and the bearing selection based on this modified load will be more realistic than a selection based on the peak torque load.

They Say...

"An outstanding characteristic of our civilization seems to be almost complete dependence upon inventions of every nature; mechanical, electrical, medical, electronic, and added to these, daily disclosures of every new discovery. Study and planning are of paramount importance if we are to continue in our social progress and our material wealth and power."—HAL F. FRUTH, *consultant and physicist, H. F. Fruth & Associates.*



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Modifying Starwheel Mechanisms

. . . to improve high-speed properties and provide versatile operation

MECHANISMS for intermittent motion are among the most useful devices available to designers. They have accordingly been the object of much investigation; perhaps the most notable recent contribution is the critical survey of intermittent mechanisms by Otto Lichtwitz.[1] Although these mechanisms have great utility, some of them are severely limited with respect to their kinematic properties which are dictated by geometry rather than the designer's wishes. Thus, Geneva and starwheel mechanisms may sometimes not be suited to applications involving high speeds or large masses or to machines requiring accuracy and smoothness of operation.

Since, as Lichtwitz points out, the most versatile of the intermittent mechanisms is the starwheel type, it will be the specific subject of discussion here on how kinematic properties can be modified and improved.

Overcoming Limitations: In designing a basic starwheel mechanism, the designer has control over only one factor—the ratio of the pitch radii of the two uniform-motion portion gears. This ratio is selected to give either a certain desired velocity or a certain time interval for indexing through a certain distance.

Choice of this single factor for the basic starwheel also sets the kinematic characteristics. This simultaneous restriction results from the specification for the standard starwheel mechanism, ***Fig.*** **1*a*,** that a uniform-width slot and mating roller shall be used for the nonuniform-motion portion of the cycle. To suit exit requirements for this condition, the slot takes the path of an epicycloid which gives inherently unfavorable kinematic properties. A relatively high acceleration is instantly applied at the start of motion, or in other words, it is the equivalent of a cam having infinite "jerk" (third derivative of displacement).

The restricting effect of the epicycloid can be eliminated only if some physical change is made in the arrangement of parts to permit the choice or specifi-

[1] References are tabulated at end of article.

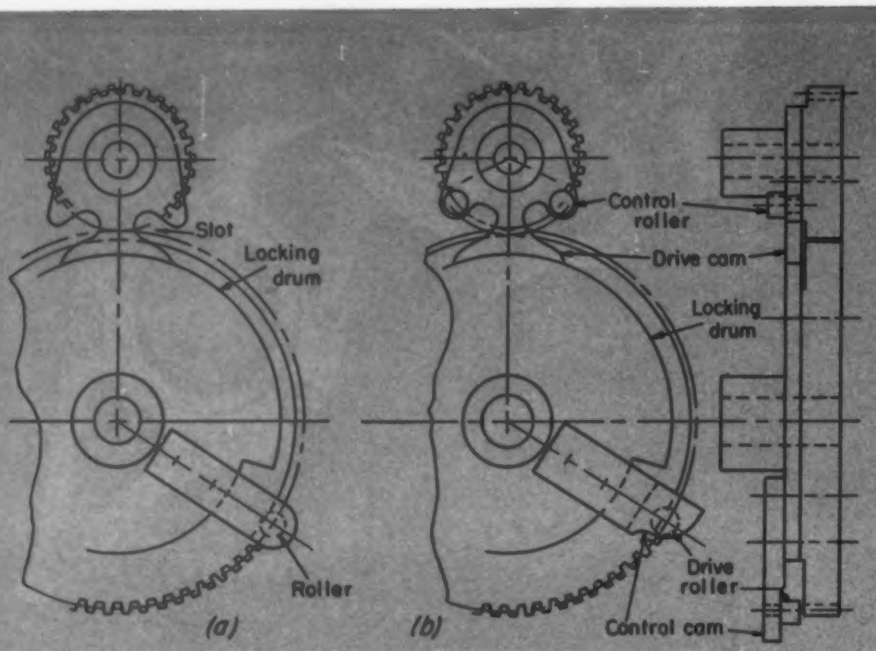


Fig. 1—Left — Basic starwheel mechanism, *a*, and modified mechanism, *b*. Roller slot of the basic starwheel is an epicycloid. In the modified starwheel, separate drive and control cams with their respective rollers permit selection of any transition function

Fig. 2—Right — Unidirectional - stopping starwheel applied to the drive of a printing press

cation of different cam-like surfaces. Such modification has been made by Harris-Seybold on starwheel mechanisms employed in printing press drives.

The solution, *Fig.* 1*b*, is separation of the sides of the slot into two cam surfaces—one on the driving member and one on the driven member. Correspondingly, two rollers are used, one for each of the driving and driven members.

As *Fig.* 1*b* shows, the drive roller on the driving member and the drive cam on the driven member are, to all appearances, equivalent to the roller and slot of the basic starwheel, *Fig.* 1*a*. However, the drive roller engages in the driven member on the drive cam and has clearance with the other side of the slot. Positive engagement, therefore, is the only function of the added control roller on the driven member and the control cam on the driving member.

With these provisions, the cam surfaces may take almost any prescribed form. In fact, there is now the opportunity to treat the speed changing portion of the starwheel as though it were a regular cam and follower system.

Motion Patterns: Somewhat more flexible in application than the basic mechanism, the modified starwheel can more easily be employed to produce three principal patterns of motion:

1. *Unidirectional-stopping*—the standard type for use where the standstill period is required to suit the function of the machine.
2. *Unidirectional-nonstopping*—for use where a standstill is not required. The lock is eliminated, and acceleration is reduced or more uniform-motion periods can be used.
3. *Bidirectional*—for use where oscillating motions are required, such as to provide for the reciprocation of another member.

Although simple indexing problems are not included in this discussion, such applications can also be treated by the methods outlined here if only a few stations are involved.

Motion Pattern Applications: Examples of all three motion patterns are found among the various lithographic offset and typographic presses manufactured by Harris-Seybold.

A modified starwheel for unidirectional-stopping action is shown in *Fig.* 2. Here the mechanism drives a 12-inch cylinder weighing approximately 1000 pounds at an average velocity of 100 rpm with one standstill period for each revolution. The velocity of this cylinder during its uniform-motion period is 175 rpm. No wear of the cam surfaces has been noted after three years of operation.

The bidirectional pattern is applied in another press for driving a reciprocating 185-pound carriage at about 110 cycles per minute. Travel is at a uniform rate for 12 inches; stopping and starting during reversal occupy an additional 3 inches at each end. Design and operation of the bidirectional mechanism are depicted in *Fig.* 3. Two identical drivers rotate in opposite directions and alternately drive a single driven member by means of separate cams and rollers.

Each of the acceleration and deceleration periods occupies 45 degrees of driver rotation. Single-crank mechanisms used for reciprocation provide only 30 degrees for accelerations or decelerations which are of inversely greater magnitude. With more complicated and cumbersome double-crank mechanisms, a 45-degree acceleration period can be obtained but kinematic properties cannot be favorably manipulated. Such crank mechanisms also take up more space than a starwheel.

Where an intermittent motion is desired but stopping is not necessary, the modified unidirectional nonstopping starwheel is especially appropriate. Advantage is gained, of course, because the change in velocity is reduced. That is, acceleration is reduced if

STARWHEEL MECHANISMS

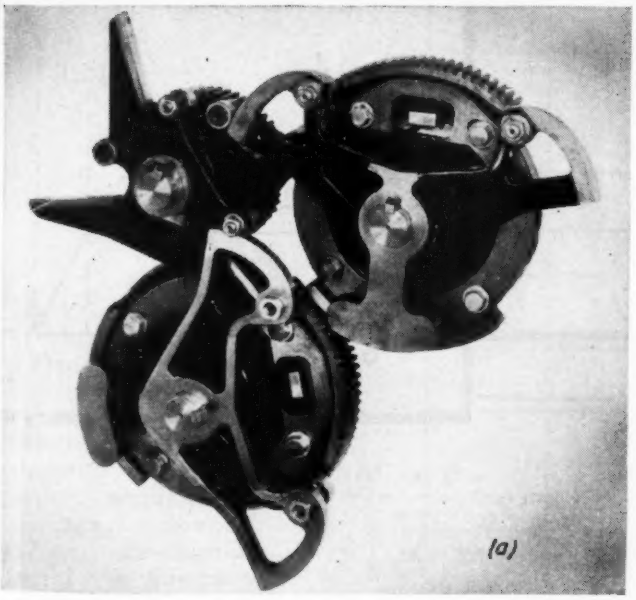
the period of time allowed for the operation remains the same. The nonstopping design also eliminates the locking device normally required with mechanisms having standstill periods. Such a design for a press is shown in *Fig.* 4. In this case, the system can be thought of as a two speed transmission with an automatic shift of a positive order being provided by the starwheel.

A fact giving an added note on the practical application of these mechanisms is that a tolerance of ± 0.001 inch is maintained in the register of sheets handled by the starwheel-driven cylinders. Some features of these mechanisms, incidentally, are the subjects of patents assigned to Harris-Seybold.

Limitations and Advantages: By comparison with standard starwheels, those of modified design require consideration of certain restricting conditions:

1. Just as for standard starwheels, a minor problem is provision of clearance between cams and rollers during uniform-motion periods. This factor does not necessarily limit the form of the cam but it may curtail the period allowable for cam and roller action.
2. Radii swept by both components of the modified drive are greater than those required by the basic starwheel.
3. Number of stations possible is limited by the increased time and space required by the nonuniform motion action and components.

The outstanding advantage of the modified starwheel is the designer's opportunity to select a cam



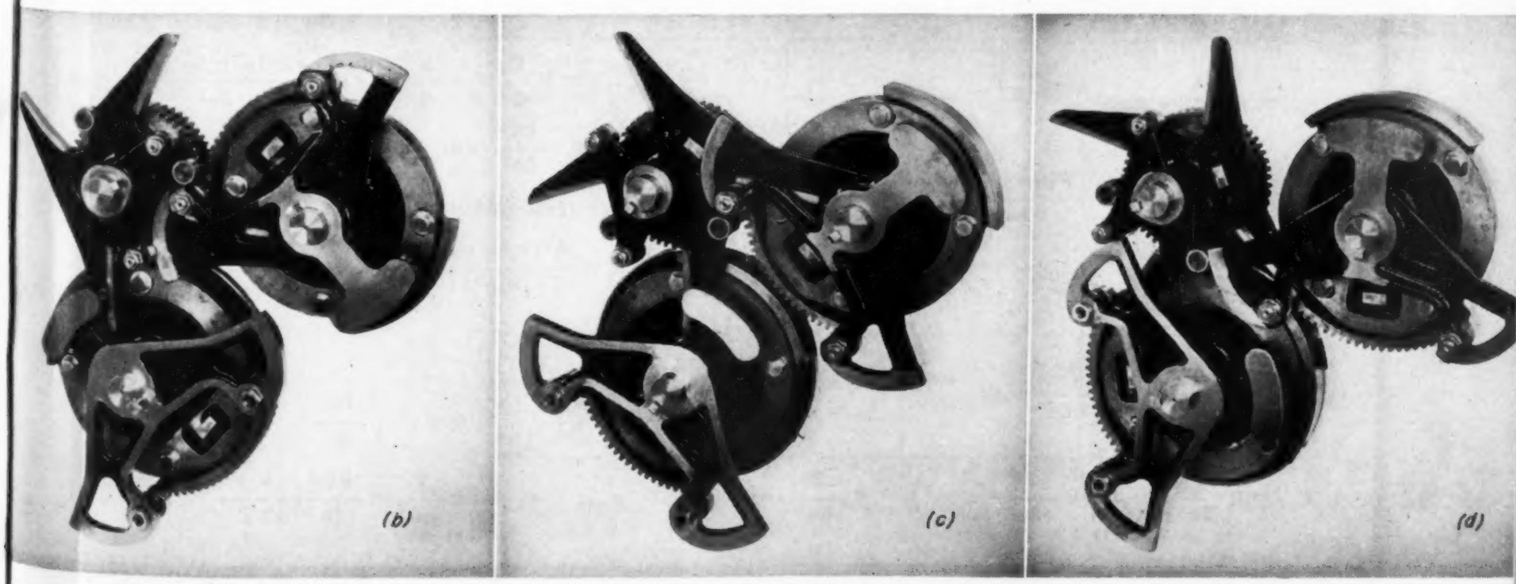


Fig. 3—Bidirectional starwheel mechanism for providing reciprocating motion. The two identical members rotate continuously in opposite directions, alternately driving the driven member. Sequence photos show transition from: *a*, reverse deceleration to forward acceleration; *b*, forward acceleration to forward uniform speed; *c*, forward uniform speed to forward deceleration; *d*, forward deceleration to reverse acceleration

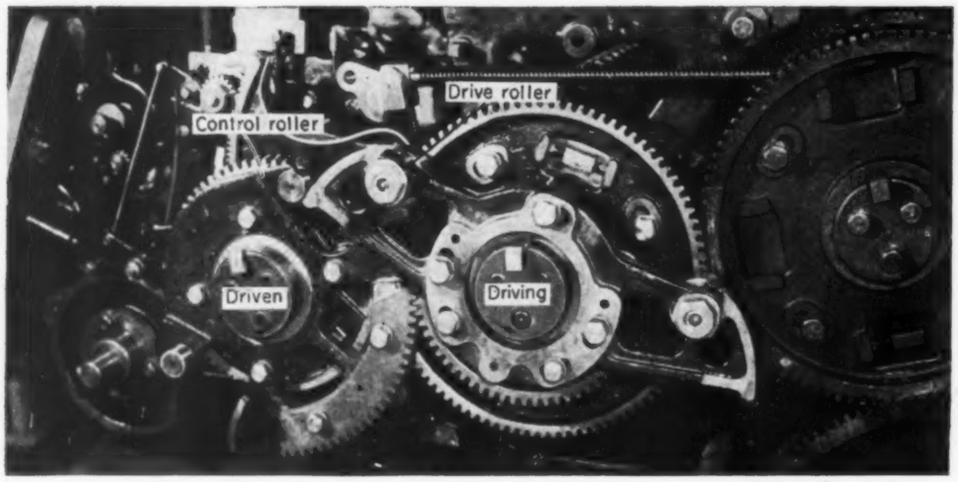


Fig. 4—Unidirectional nonstopping starwheel mechanism for slowing instead of stopping the driven member. The cams and rollers provide, in effect, an automatic positive shift between two uniform-velocity stages

function that gives a desired result rather than to be forced to accept a constricting condition. Being able to control the manner in which velocity is altered, the designer can minimize acceleration and thus inertia loads. The process, of course, involves selection of the desired acceleration characteristics and design of the cam profiles by integration of the acceleration function.

Design: With kinematic objectives in sight, instead of only the geometry of action, elements to be considered in design are:

1. Time that can be allotted to the acceleration portion of the cycle.
2. Velocities desired at both ends of the acceleration period.

Fig. 5 — Trapezoidal acceleration diagram employed with the modified star wheel designs

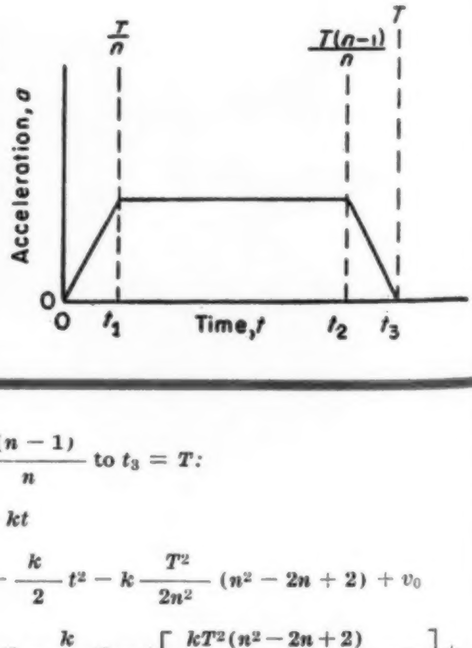


Table 1—Equations for Trapezoidal Profile

From $t_0 = 0$, to $t_1 = \dfrac{T}{n}$ (see *Fig.* 5):

$$a_{0\text{-}1} = kt$$

$$v_{0\text{-}1} = k\frac{t^2}{2} + v_0$$

$$s_{0\text{-}1} = k\frac{t^3}{6} + v_0 t + s_0$$

From $t_1 = \dfrac{T}{n}$ to $t_2 = \dfrac{T(n-1)}{n}$:

$$a_{1\text{-}2} = k\frac{T}{n}$$

$$v_{1\text{-}2} = k\frac{T}{n}t - k\frac{T^2}{2n^2} + v_0$$

$$s_{1\text{-}2} = k\frac{T}{2n}t^2 - t\left(k\frac{T^2}{2n^2} - v_0\right) + k\frac{T^3}{6n^3} + s_0$$

From $t_2 = \dfrac{T(n-1)}{n}$ to $t_3 = T$:

$$a_{2\text{-}3} = kT - kt$$

$$v_{2\text{-}3} = kTt - \frac{k}{2}t^2 - k\frac{T^2}{2n^2}(n^2 - 2n + 2) + v_0$$

$$s_{2\text{-}3} = k\frac{T}{2}t^2 - \frac{k}{6}t^3 - t\left[\frac{kT^2(n^2-2n+2)}{2n^2} - v_0\right] + \frac{kT^3}{6n^2}(n^2 - 3n + 3) + s_0$$

Other relationships:

$$k = \frac{n^2(v_3 - v_0)}{T^2(n-1)}$$

$$s_3 = \frac{T}{2}(v_3 + v_0)$$

$$\left(\frac{d^3s}{dt^3}\right)_{0\text{-}1} = k = -\left(\frac{d^3s}{dt^3}\right)_{2\text{-}3}$$

$$a_{max} = a_{1\text{-}2} = k\frac{T}{n} = \frac{n(v_3 - v_0)}{(n-1)T}$$

3. Displacement range over which the acceleration may occur.
4. Minimum peak acceleration.
5. Minimum peak jerk[2] with due consideration for its effect on peak acceleration and for inherent susceptibility of the machine to vibration.

Any desired acceleration function may be used, but for the purposes of this discussion a trapezoidal acceleration curve of the type shown in *Fig.* 5 will be developed and applied to several examples. This form has been adopted in recent designs by Harris-Seybold because the trapezoidal curve (1) approaches the constant-acceleration type profile in yielding low peak acceleration, and (2) provides finite jerk throughout the range of cam action. This approach has been further substantiated by Neklutin.[3]

As *Fig.* 5 shows, t represents any time within the event, T, the total time period. The fractional parts of T marking the four points at which the jerk, k, changes (t_0, t_1, t_2, t_3) are controlled by n, an arbitrary factor which can be altered to change the value of peak acceleration. Displacement shall be represented by s, velocity by v, and constant jerk (the sloped regions of the acceleration diagram) by k. Equations for the three portions of the acceleration diagram, *Fig.* 5, can be easily developed, and then successively integrated to give velocity and displacement. The results are shown in TABLE 1 and are applicable, of course, to a variety of cam types.

Comparisons: The starwheel shown in *Fig.* 2 serves well as the basis for a number of comparative inves-

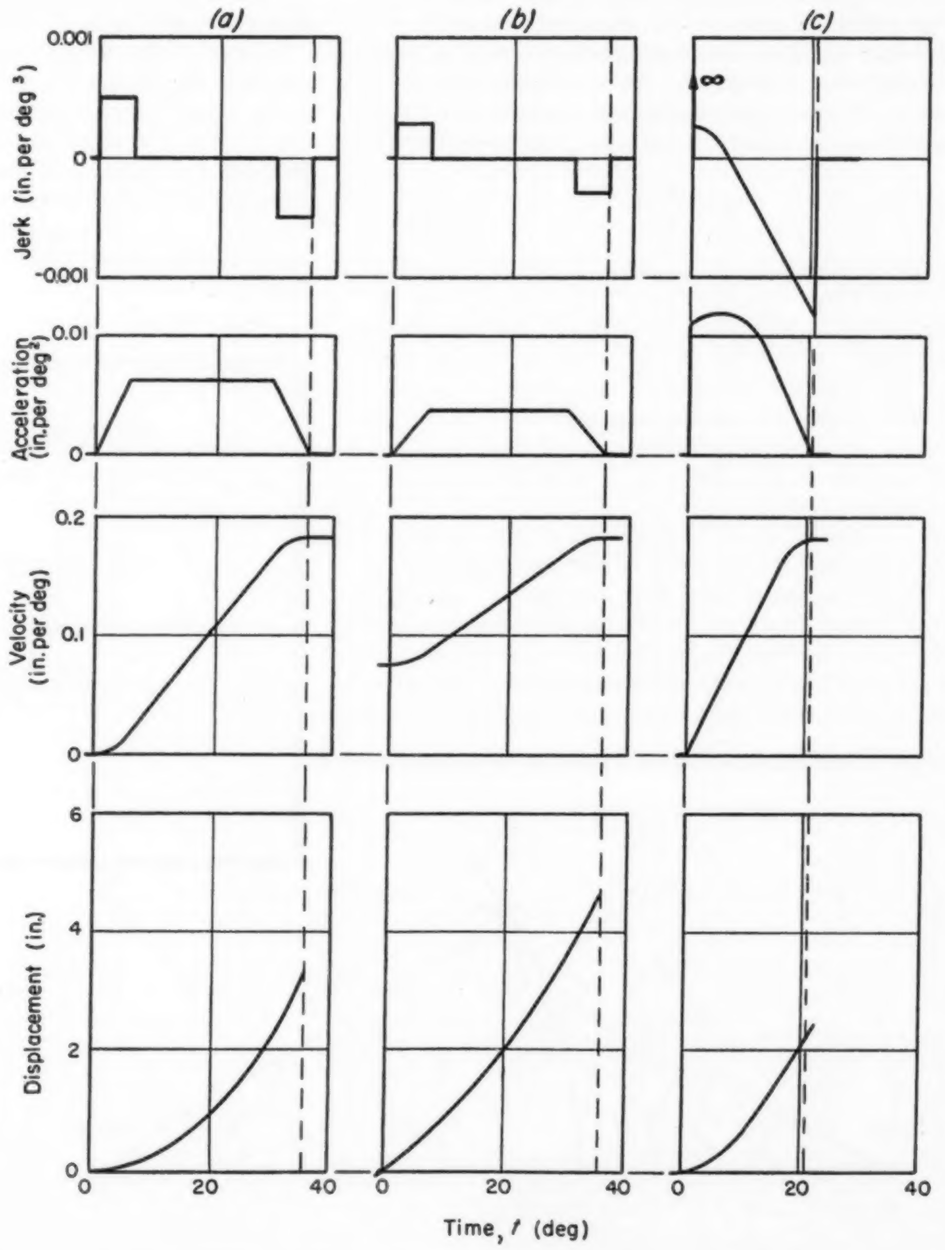


Fig. 6 — Comparison of characteristics for: *a*, modified unidirectional stopping mechanism; *b*, modified unidirectional nonstopping design; *c*, basic unidirectional stopping starwheel. Acceleration diagrams reveal the advantage of the modified starwheels, particularly of the nonstopping design at *b*

tigations. This mechanism consists of a 21-inch diameter driving member and a 12-inch driven member. For the first calculation, the driven cylinder shall be accelerated from standstill, $v_0 = 0$, to the surface velocity of the driving cylinder. Or, with degrees used as the time base, $v_3 = 21\pi/360 = 0.1833$ in. per deg. The total period shall be assumed to be $T = 36$ deg, and the divisions of the event shall be apportioned according to $n = 6$.

Then, by the equations in TABLE 1, points can be calculated and the curves in *Fig.* 6*a* plotted. Also from TABLE 1, $k = 0.001018$ in. per deg^3, $a_{max} = 0.006108$ in. per deg^2, and $s_3 = 3.298$ inches or 31.49 deg.

The same mechanism, except that it shall be of the nonstopping type, can next be investigated. Assume that $v_0 = 0.40\ v_3 = 0.40\ (0.1833) = 0.0733$ in. per deg. Plotted results are shown in *Fig.* 6*b*, and the comparative data are $k = 0.000611$ in. per deg^3, $s_3 = 4.619$ inches, and $a_{max} = 0.00367$ in. per deg^2. Acceleration for the nonstopping mechanism, thus, is 40 per cent less than that for the standstill design.

These two designs are next compared with a corresponding basic starwheel. The complete index to properties of the basic starwheel, according to the nomenclature employed by Lichtwitz, is provided by

$$\mu = \frac{r_2}{r_1}$$

By use of equations or the table of values presented by Lichtwitz for external starwheels[1], results can be readily obtained in units of radians and seconds. These results, converted to the units used here, are plotted in *Fig.* 6*c*. Pertinent comparative data in terms of the two nomenclatures, are: $T = \alpha_0 = 21$ deg, $s_3 = \beta_0 = 2.29$ inches, $a_0 = (d^2\beta/d\alpha^2)_{-\alpha 0} = 0.0106$ in. per deg^2, and $a_{max} = (d^2\beta/d\alpha^2)_{max} = 0.0117$ in. per deg^2.

These examples show that the maximum acceleration for the basic starwheel is 92 per cent higher than that for the modified stopping mechanism, and 219 per cent higher than that for the modified nonstopping system. The designs might also be compared on another basis. If all these mechanisms were to drive the same member with a given maximum torque, the driving speeds possible with the three mechanisms would be comparatively as follows: For the modified stopping design, 78; for the modified nonstopping design, 100; and for the basic starwheel, 56.

Cam Geometry: Foregoing discussion has been almost entirely confined to the kinematics of the subject, but execution of the profiles that will give such improved results remains now as a simple problem in geometry.

Polar or rectangular co-ordinates are usually the best means of defining cam contours that are to be step milled—the method employed in the cutting of the drive and control cams used in starwheel mechanisms. Since the diameter of the milling cutter is normally the same diameter as the follower roller, the path of the roller center (or cutter center) is usually defined by the co-ordinates. The method employed at Harris-Seybold for calculating polar co-ordinates of this path—the pitch surface of the cam—is summarized here.

In *Fig.* 7, the initial position and a later displaced position are shown for the driving and driven members. Path of the driving roller is arc *AD'* about center *B*. Its rotational speed is constant, and its position *t* is measured from starting line *AB*, which thus is t_0. The total arc of action is β_0; if "time" is based on rotational angle, $\beta_0 = T$. Factors *t* and *T* have already been defined in reference to *Fig.* 5 and TABLE 1.

Table 2—Calculation of Drive Cam

Polar co-ordinates (see *Fig.* 7):

$$\alpha_2 = \rho + s'$$

$$l_2 = \sqrt{r_1^2 + a^2 - 2r_1\, a \cos\beta}$$

Auxiliary relationships:

$$\rho = \tan^{-1}\frac{r_1 \sin\beta}{a - r_1\cos\beta}$$

$$\beta = T - t$$

$$a = r_1 + r_2, \qquad s' = \frac{180}{\pi}\,\frac{s}{r_2}$$

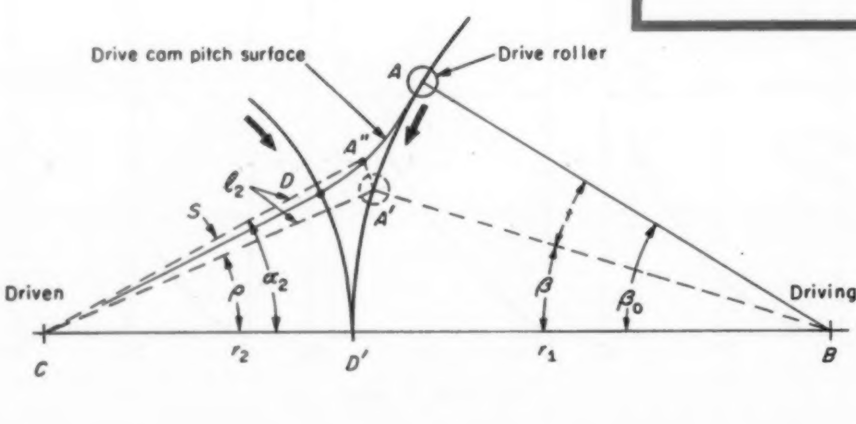


Fig. 7—Geometry of the drive cam

As the drive roller moves from position *A* to *A'*, the cam pitch surface provides displacement angle *s'* of the driven member. Correspondingly, the contact of the roller center moves from *A* (the initial effective point of the cam pitch surface) along the pitch surface to point *A''*. The terminus of action is point *D* on the driven pitch surface, and point *D'* is the final location of the drive roller.

Polar co-ordinates of point *A''* on the driven pitch surface are angle α_2 and radius l_2. Equations for the co-ordinates and required supplementary relationships are given in TABLE 2.

Profile of the control cam is developed according to *Fig.* 8 and TABLE 3. Although components in *Fig.* 8 have the same relative placement as in *Fig.* 7, action is most conveniently depicted if the driving mechanism is considered to be rotating about center *C* of the driven member. That is, for an increment of displacement, center *B* of the driving member revolves about point *C* to point *B'*.

Angle ϕ_0 is arbitrarily selected, although choice is influenced by clearance required between control and drive rollers when the drive roller is situated at *D*. In action, as the center of rotation of the driving member moves from *B* to *B'*, equivalent to rotation of driving and driven members, the control roller center, in effect, generates the pitch surface of the control cam according to the displacement function already dictated by design of the drive elements.

As shown in *Fig.* 8, the control cam pitch surface moves from its initial position to that noted by the dashed construction. The properties that serve as co-ordinates for the location of point *E* on the control cam pitch surface are angle α_1 and radius l_1. Angle α_1 has a starting or initial value $\alpha_{1\text{-}0}$ with its apex at origin *B;* its value α_1 after rotation is measured from origin *B'*, etc.

"Timing" of the drive and control cam surface is automatically controlled by the use of co-ordinate data from the drive cam in the calculations for the control cam.

In the step milling of cam profiles, cuts should be closely enough spaced to permit accurate results with minimum filing or hand dressing between cuts. Peripheral increments of 0.08 to 0.10-inch have proved satisfactory in the range of sizes shown by the foregoing examples.

Other Possibilities: With acceptance of the idea that mechanisms such as starwheels need not be restricted in design to their classic forms, a number of worthwhile possibilities become available. Design for altered kinematic properties can take almost any path. Even the recently announced "polydyne" principles[4] can be applied if circumstances warrant.

Also, certain previous geometric limitations can be modified so that various hybrid designs can be developed. That is, several of the motion patterns discussed here might be combined in a single design. These hybrid or integrated designs are worthwhile because they simultaneously eliminate other auxiliary mechanism systems, and often provide improved control of inertia loads.

Table 3—Calculation of Control Cam

Polar co-ordinates (see *Fig.* 8):

$$\alpha_1 = \beta - \theta$$

$$l_1 = \frac{r_2 \sin \phi}{\sin \phi}$$

Auxiliary relationships (also see TABLE 2):

$$\theta = \tan^{-1} \frac{r_2 \sin \phi}{a - r_2 \cos \phi}$$

$$\phi = \phi_0 - \alpha_2 + \rho$$

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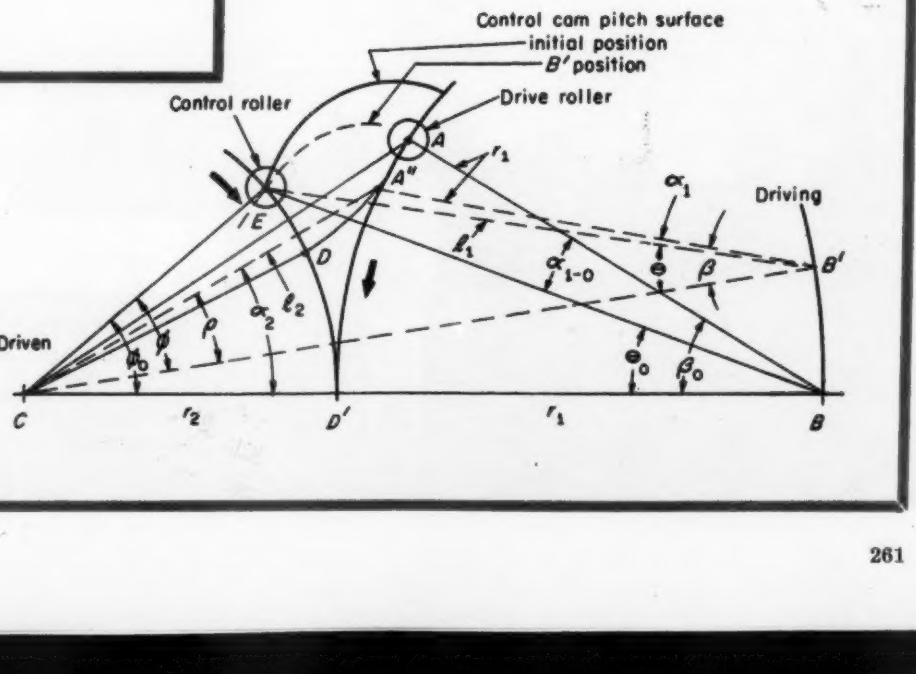


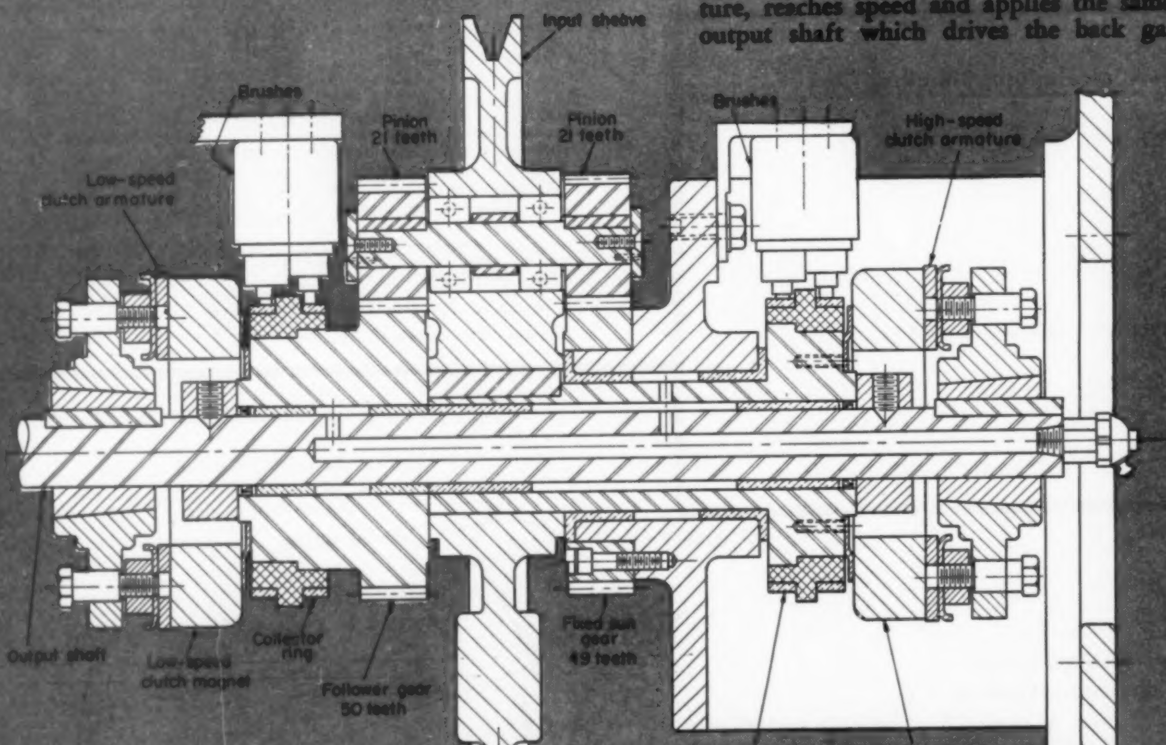
Fig. 8—Geometry of the control cam

AUTOM

Harris-Seybold auto-spacer cutter designed for repetitive cutting of paper stock. Back gage of machine advances the stack to successive preselected positions for cutting. Forward traverse is at high speed over most of the required distance, low speed for final positioning

←

↓ Auto-spacer drive unit showing gearing and electric clutches. Both clutch armatures rotate continuously. High-speed armature is direct-connected to the input sheave; low-speed armature is connected to the input sheave through the planetary gearing which provides a speed reduction of 285/6. When either clutch magnet is energized, it comes into contact with its armature, reaches speed and applies the same speed to the output shaft which drives the back gage lead screw



OMATIC SPACING CONTROL

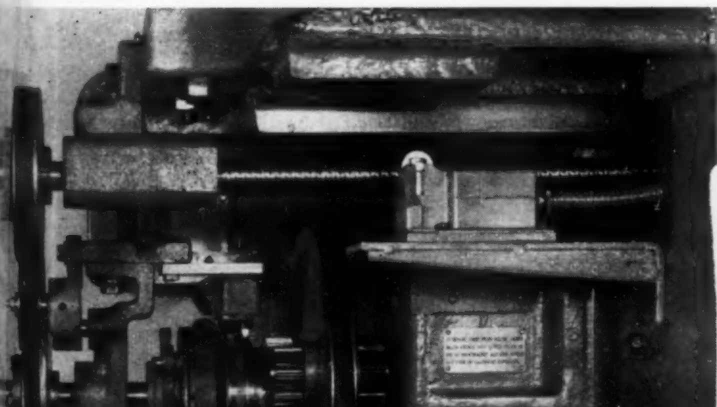
. . . for paper cutter provides high accuracy at low cost with electric clutches and planetary gearing

IMPROVED performance and lower cost are design objectives that are all the more worthwhile if they can be realized simultaneously. Both these objectives were met in the drive and control system for the automatic spacing feature of the Harris-Seybold paper cutter discussed in this article. The design illustrates well how modern mechanical and electrical components can be coupled through a comprehensive control system to produce a sound design.

The purpose of the auto-spacer facility is to provide a certain degree of automaticity in the repetitive cutting of paper products. The stack of paper to be cut is loaded on the table to the rear of the machine and pushed into cutting position by a lead-screw driven back gage. Speed of advance of the back gage is sharply reduced as the stack approaches the cutting position, and advance is terminated automatically when the preselected position is reached. Descent of the knife, powered by a separate motor, is manually controlled by the operator. The control system also provides for pushbutton operation of the back gage on two speeds forward and one speed reverse for short jobs set up for cutting to a line rather than for automatic spacing of cuts.

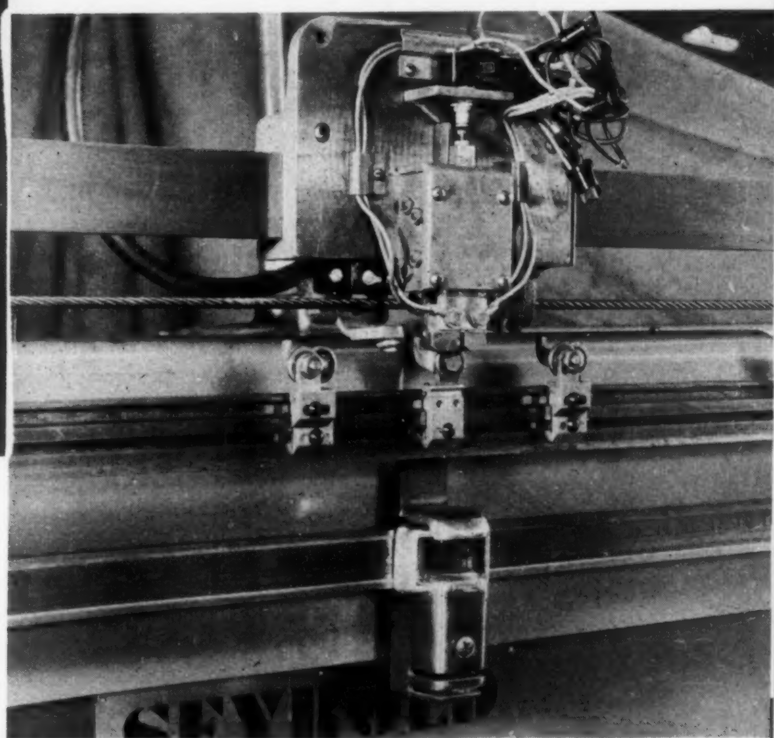
In earlier models the high-speed traverse was provided through spiral-bevel reduction gearing; low speed was provided by a chain-driven worm-reduction pair. Speed shift was controlled by a solenoid-operated jaw clutch. Replacement of this design by the system depicted in the accompanying illustrations proved beneficial in the following respects: increased speed of operation, elimination of noise, greater accuracy of positioning, and reduction in cost.

Both the drive and control phases of the new model reflect sound engineering. The planetary gearing yields a high reduction in small space with relatively



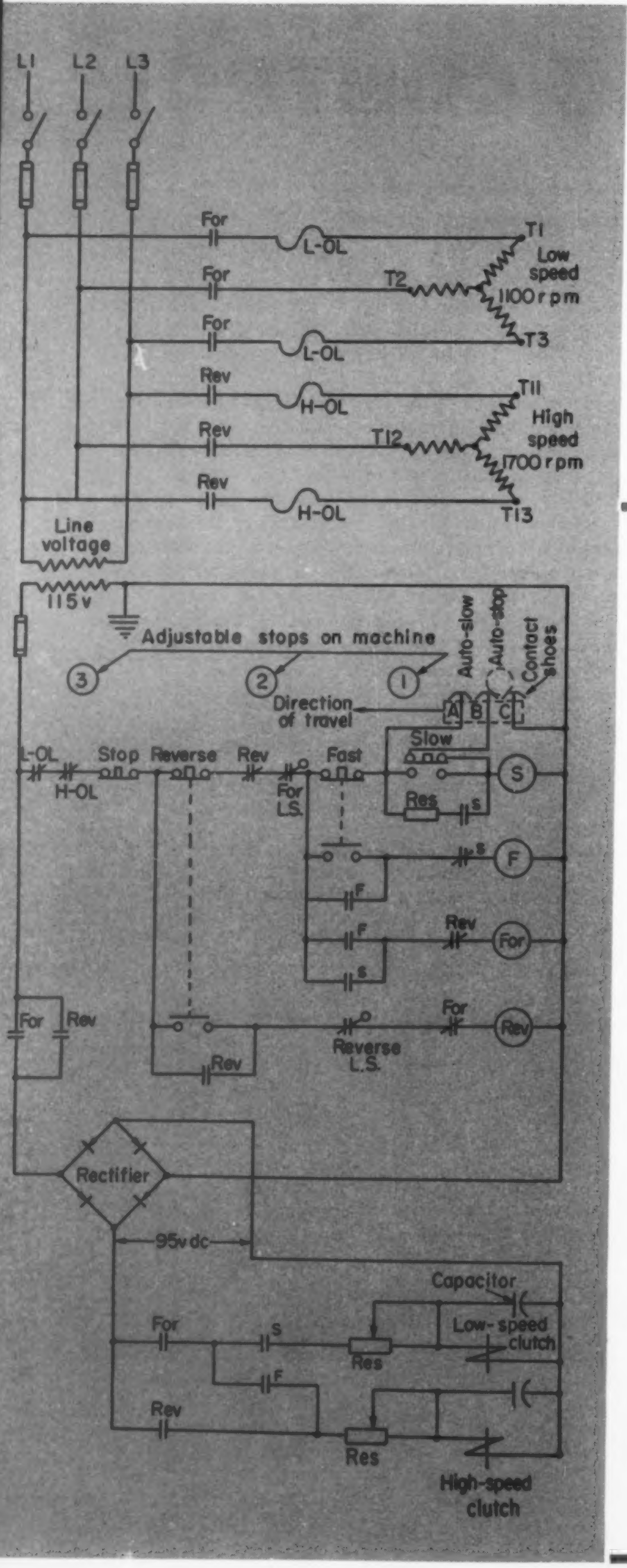
Part of the drive unit showing the planetary gearing, the high-speed clutch, and the belt drive to the back gage lead screw. A two-speed reversible motor supplies 1100 rpm for high and low-speed forward travel, 1700 rpm for reverse travel through the high-speed clutch. Corresponding traverse speeds of the back gage are 28.5 fpm forward, 0.6 fpm forward, and 43 fpm reverse

Auto-spacer stop system showing contact head, which is cabled to the back gage, and the adjustable spacing stops. Housing and front roller on central stop are removed to show details. Contact established by the L-shaped member causes shift from high to low-speed traverse of the back gage. Low-speed traverse is cut off when the contact shoes are positioned on the roller. Positioning tolerance is maintained within a few thousandths of an inch



simple and inexpensive components. The electric clutches, employed at only a fraction of their rating, give rapid response, long life and nearly silent operation. The low-speed clutch, besides its drive function, also serves as a brake against the inertia during transition from high to low speed. The electromagnetic operation of the clutches is well adapted to the electric control system which provides for operational flexibility and safety.

This machine also demonstrates a principle which is a growing factor in effective engineering design—the collaboration of manufacturers of both original equipment and standard components. In this case, both Cutler Hammer Inc. and the Warner Electric Brake and Clutch Co. co-operated with Harris-Seybold in the development of the design.



Simplified diagram of control system. Details of operation are:

Automatic Control

1. Press *Fast* pushbutton. Contactor *F* is energized, holds by contact, and energizes *For* contactor. Motor low-speed winding (1100 rpm) is connected to line and motor runs at that speed forward. Contacts of *For* and of *F* energize high-speed clutch and back gage moves forward at fast rate.
2. Adjustable stop roller on machine makes circuit between contact shoes *A* and *B*. Contactor *S* is energized and holds by contact through resistor. Contactor *F* is de-energized by opening of normally closed contact of *S*. Contactor *For* remains energized through contact of *S* and motor continues to run forward, now at low speed. High-speed clutch is de-energized by opening of *F* and slow-speed clutch is energized by closing of *S*. Back gage continues forward travel at slow speed.
3. Adjustable stop roller, which caused slowdown, makes circuit between contact shoes *B* and *C*. Coil of contactor *S* is short-circuited and contactor *S* opens. Contactor *For* is de-energized and opens to disconnect the motor. Low-speed clutch is de-energized and backgage movement stops.
4. Sequence of operation through stages 1 to 3 is repeated in the forward direction for all preset stops.
5. At end of forward movement or after the last automatic stop, press *Reverse* pushbutton. Contactor *Rev* is energized and holds by contact. Motor high-speed winding (1700 rpm) is connected to line and motor runs at that speed reverse. Contact of *Rev* energizes *Fast* clutch and back gage returns at high speed.

Manual Control

Forward or reverse operation can be stopped at any time by pressing *Stop* pushbotton. Motor and clutches are immediately de-energized by opening of all contactors. Immediate shift from one speed to another, forward or reverse, is obtained by pressing the pushbutton for the desired speed.

Safety Features

Forward limit switch prevents forward overtravel of back gage. Reverse limit switch prevents reverse overtravel. Relays prevent damage to motor if overload condition arises. Control circuit is low voltage with one side of transformer secondary winding grounded

Charts Simplify Hoisting Drum Design

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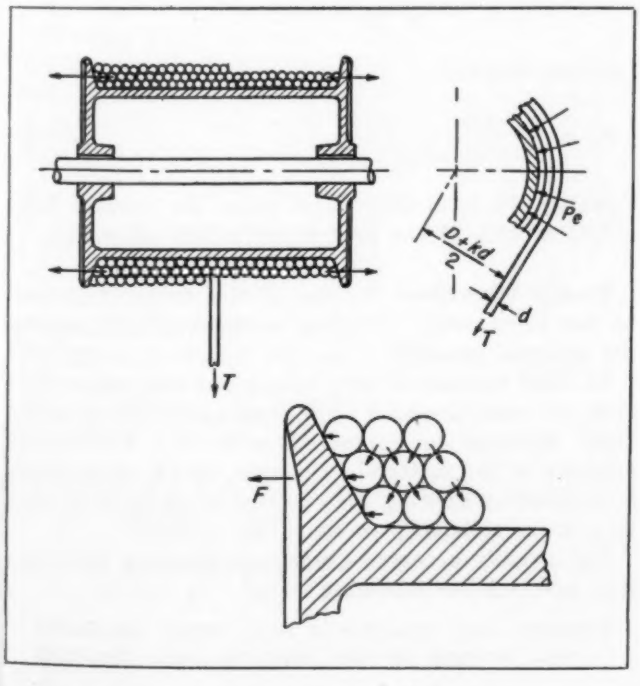
STRESS conditions in a multiple-layer grooved rope drum are complex. However, if the drum is assumed to be a cylindrical shell with uniform wall thickness, calculation of stress components is possible. Loads acting on the drum body are readily analyzed and stress components and collapsing pressure are expressed in terms of drum dimensions, load and material properties. Based on dimensionless grouping of the variables, charts can be constructed for rapid calculation. A simplified design procedure employing these charts is presented in this article.

Fig. 1—Load-analysis diagrams for hoisting drum shell showing action of axial force F, external pressure P_e, and rope tension T



Drum Loads: From an analysis of Fig. 1, stresses induced in the hoisting drum shell are the result of loads imposed by:

1. External pressure (P_e) from coiling of ropes under tension.
2. Axial force (F) in multiple-layer drums from lateral pressure on the drum flanges caused by the pressure of the outer rope layers on the inner layers.
3. Bending moment from rope tension.
4. Torsional moment from rope tension.

External pressure can be calculated from

$$P_e = \frac{2\,KT}{p\,D} \qquad (1)$$

Symbols are defined in the Nomenclature and *Figs.* 1, 2, and 3. In Equation 1 K is a constant dependent on the number of rope layers and has the values given in TABLE 1. The outer rope coils exert pressure on the inner coils and cause relaxation of part of the inner tension. Thus, the total tension for the purpose of pressure calculation is KT in lieu of nT.[1]

The magnitudes of axial force, maximum bending moment and maximum torque depend on the drum and rope arrangement. Axial force can be generally expressed by

$$F = CKT \qquad (2)$$

where C is a constant dependent on the arrangement of rope layers on a multiple-layer rope drum. For single layer drums $C = 0$, while from TABLE 1, $K = 1$. For more than one layer, the method described by Waters[1] can be used.

The maximum bending moment due to rope tension can be established from the drum layout, which is shown in *Fig.* 2 for two of the most frequently used

[1] References are tabulated at end of article.

Table 1—Values of Constants

Number of Rope Layers n	Constants k*	K
1	1	1
2	2.7	1.75
3	4.4	2.0
4	6.1	2.25

*Values given refer to close-coiled ropes; however, the error for grooved drums is on the safe side.

arrangements. In *Fig. 2a,*

$$M_b = \frac{Tl}{4} \qquad (3)$$

and in *Fig. 2b,*

$$M_b = Tl_o \qquad (4)$$

The additional bending moment caused by the distributed weight of the drum is usually small compared to the maximum and can be neglected.

Maximum torsional moment for the arrangement in *Fig. 2a* is

$$M_t = \frac{T(D + kd)}{2} \qquad (5)$$

In *Fig. 2b,*

$$M_t = T(D + kd) \qquad (6)$$

where k is a function of the number of rope layers and has the values listed in TABLE 1.

Stress Components and Collapsing Pressure: From *Fig.* 3, based on the assumption that the maximum bending and twisting moments occur at the same section, the stresses on small elements of the drum shell at points o and i on the outside and inside surfaces, respectively, can be expressed as follows.

Axial stress is constant across the section, giving

$$s_a = \frac{CKT}{\frac{\pi}{4}[D^2 - (D - 2t)^2]} = \frac{CKT}{\pi D^2}\left[\frac{1}{\frac{t}{D}\left(1 - \frac{t}{D}\right)}\right] \qquad (7)$$

Tangential compressive stresses are

$$s_{ci} = -\frac{2P_e D^2}{D^2 - \left(D - 2\frac{t}{D}\right)^2} = -\frac{P_e}{2}\left[\frac{1}{\frac{t}{D}\left(1 - \frac{t}{D}\right)}\right] \qquad (8)$$

$$s_{co} = \frac{s_{ci}}{2}\left[1 + \left(1 - \frac{2t}{D}\right)^2\right] \qquad (9)$$

For the radial stresses, $s_{ri} = 0$ and

$$s_{ro} = -P_e \qquad (10)$$

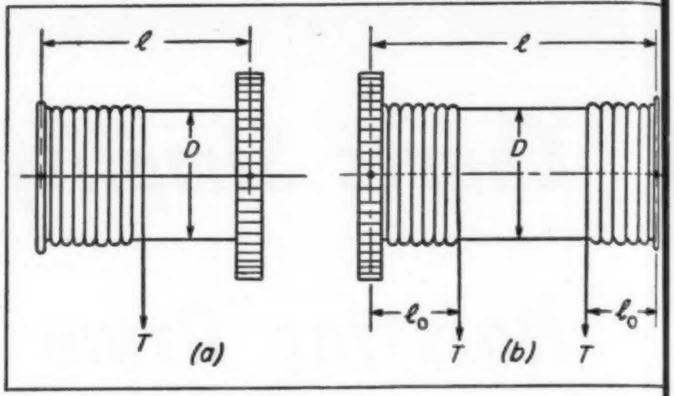


Fig. 2—Layouts for determining bending and torsional moments of the two typical drum and rope systems

Bending stresses are

$$s_{bo} = \frac{M_b}{Z_b} = \frac{M_b}{\frac{\pi D^3}{32}\left[1 - \left(1 - 2\frac{t}{D}\right)^4\right]} \qquad (11)$$

$$s_{bi} = s_{bo}\left(1 - \frac{2t}{D}\right) \qquad (12)$$

Torsional shear stresses are

$$s_{so} = \frac{M_t}{Z_t} = \frac{M_t}{2Z_b} \qquad (13)$$

$$s_{si} = s_{so}\left(1 - \frac{2t}{D}\right) \qquad (14)$$

Collapsing pressure is a function of the elastic modulus E and the ratios t/D and l/D. For intermediate lengths the following equation gives good agreement with experimental[2] results:

$$P_c = \frac{2.6E\sqrt{\left(\frac{t}{D}\right)^5}}{\frac{l}{D} - 0.45\sqrt{\frac{t}{D}}} \qquad (15)$$

For long lengths:

$$P_c = \frac{2E}{1.27(1 - \mu^2)}\left(\frac{t}{D}\right)^3 \qquad (16)$$

Equation 16 is a theoretical value for perfect long cylinders with 27 per cent imperfection allowance.

Design Procedure: Failure of the drum shell may be due to excessive combined stresses or collapse under external pressure.

To limit stresses to safe values, suitable failure criteria are used in which some combination of the principal stresses is compared with the mechanical strength of the material. Collapse, on the other hand, is avoided by limiting the external pressure to a fraction of the collapsing pressure: $P_e = P_c/N_c$

The usually accepted practice in hoisting drum design involves the following steps:

1. Select rope diameter d from tables comparing rope strength to rope diameter, rope flexibility

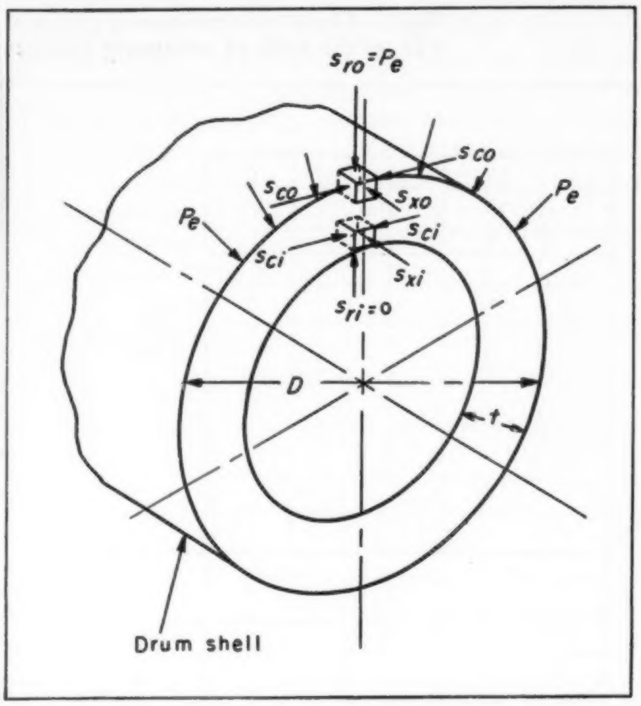


Fig. 3—Free-body diagrams for deriving stresses at inner and outer surfaces of drum shell

and wire material for a breaking load of *NT* pounds. The factor *N* usually takes care of static, acceleration and bending loads.

2. Determine the drum diameter *D* as a multiple of *d* according to the strength and flexibility of rope.
3. Determine pitch of winding *p*.
4. Determine length of drum *l* from the required length of rope, pitch and layout.
5. Determine *k* and *K* from TABLE 1 for the desired number of layers *n*.
6. Determine the maximum bending moment M_b and the maximum torque M_t from the drum layout and rope arrangement.

Thus, for an established layout *d, T, D, L, k, K, C,* M_t, and M_b are known and the design is completed by establishing a suitable wall thickness *t* for the drum. With the aid of *Figs.* 4, 5, and 6 the preliminary estimate of *t* and the calculation of stress components and collapsing pressure are greatly simplified.

In *Fig.* 4, values have been plotted for

$$A = \frac{1}{\frac{t}{D}\left(1 - \frac{t}{D}\right)} \qquad (17)$$

as a function of t/D.

The chart in *Fig.* 5 provides values of

$$P = \frac{P_c\,(10^6)}{E} = \frac{N_c\,P_e\,(10^6)}{E} \qquad (18)$$

as a function of t/D and l/D, assuming $\mu = 0.3$ for steel and cast iron. Thus,

$$P = \frac{2.6(10^6)\sqrt{\left(\frac{t}{D}\right)^5}}{\frac{l}{D} - 0.45\sqrt{\frac{t}{D}}} \qquad (19)$$

for

$$\frac{l}{D} \leqq \frac{1.5 + 0.45\left(\frac{t}{D}\right)}{\sqrt{\frac{t}{D}}}$$

and

$$P = \frac{2\ (10^6)}{1.27\,(1-\mu^2)}\left(\frac{t}{D}\right)^3 \qquad (20)$$

for

$$\frac{l}{D} \geqq \frac{1.5 + 0.45\left(\frac{t}{D}\right)}{\sqrt{\frac{t}{D}}}$$

Finally, *Fig.* 6 facilitates the calculation of the sectional modulus Z_b, providing values for

$$Z_b = \frac{\pi D^3}{32}\left[1 - \left(1 - \frac{2t}{D}\right)^4\right] \qquad (21)$$

as a function of t/D and *D*.

The following procedure, which uses the maximum stress theory for cast iron and the maximum shear theory for mild steel rope drums, is recommended. Tangential compressive stress caused by P_e is invariably the numerically largest stress; hence, a first selection of *t* for strength can be obtained by calculating

$$A = \frac{2\,s_d}{P_e} \qquad (22)$$

and obtaining the corresponding $t/D = \alpha_1$ ratio from *Fig.* 4. Also, from *Fig.* 5 another value, $t/D = \alpha_2$, is obtained which corresponds to *P* and l/D, as the criterion for collapse. Using the larger α value of

Fig. 4—Chart for determining values of t/D on the basis of strength

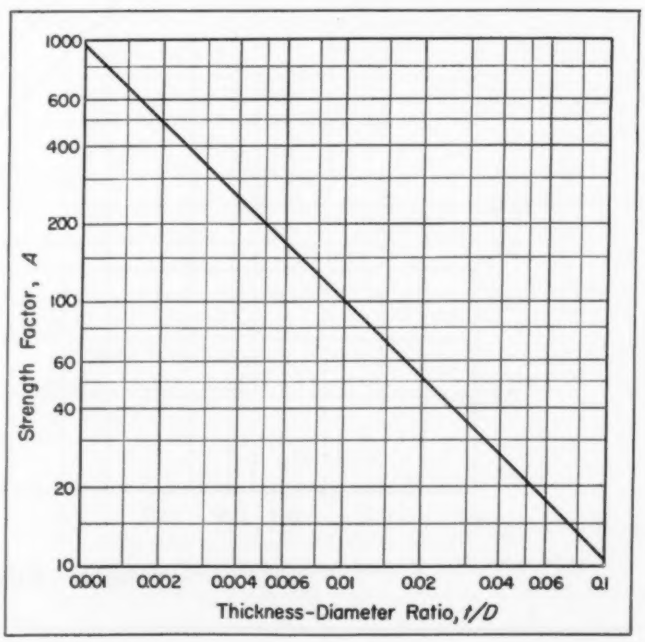
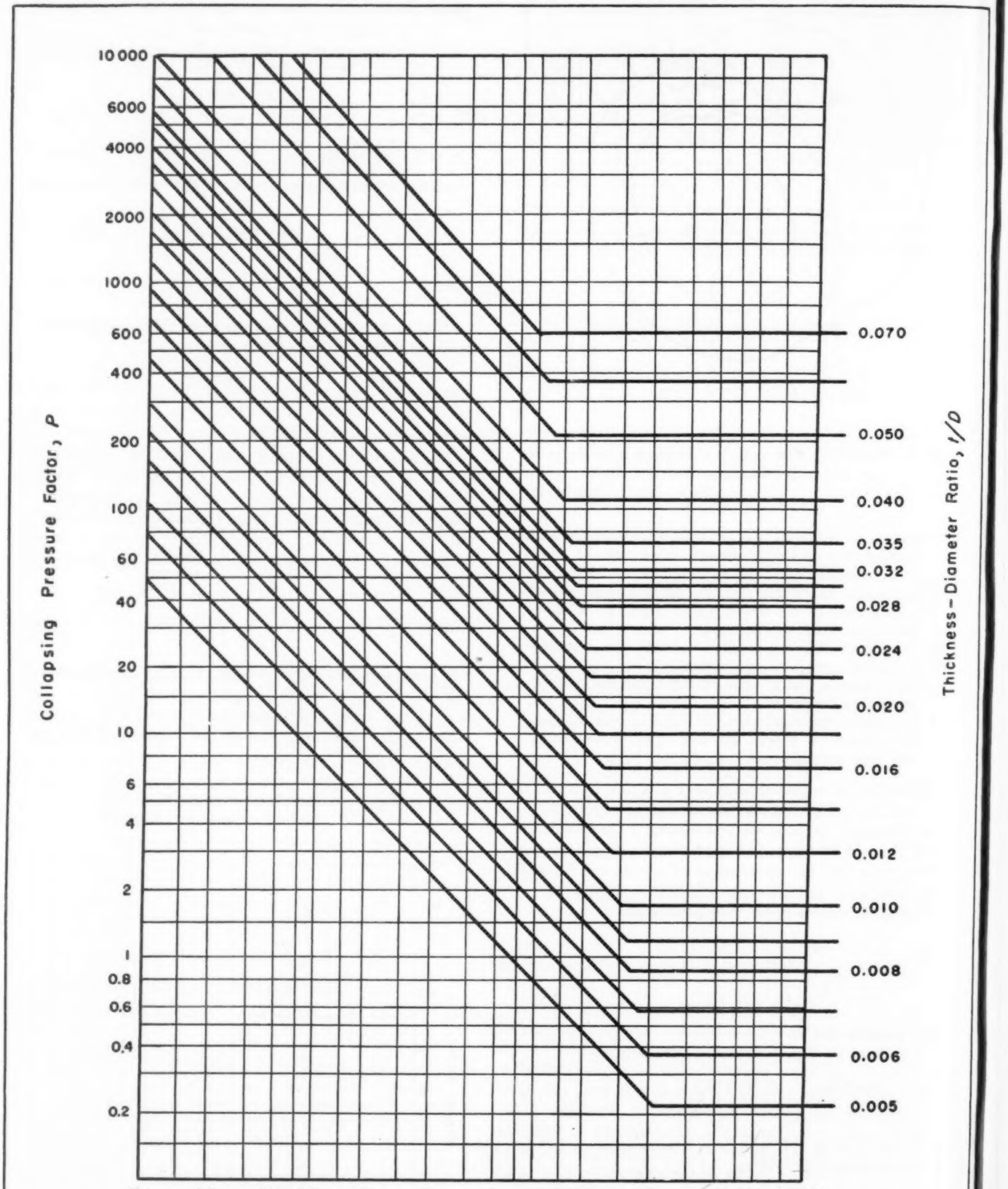


Fig. 5—Chart for determining values of t/D on the basis of collapsing pressure



the two a value for the wall thickness, $t > D\alpha$, is selected.

To simplify calculations, the value of s_{ci} is used for s_c instead of s_{co} and the stress components at the highest stressed points are calculated. The t/D ratio is calculated first from the previously selected value of the wall thickness. Then from *Fig.* 4, a value A_1 corresponding to this t/D ratio is obtained. Values for Z_b and $Z_t = 2\ Z_b$ are obtained from *Fig.* 6. The stress components thus are:

$$s_a = \frac{CKTA_1}{\pi D^2} \qquad (23)$$

$$s_c = -\frac{P_e A_1}{2} \qquad (24)$$

$$s_r = -P_e \qquad (25)$$

$$s_b = \pm \frac{M_b}{Z_b} \qquad (26)$$

$$s_s = \frac{M_t}{Z_t} \qquad (27)$$

Total normal stress in the axial direction is

$$s_x = s_a + s_b \qquad (28)$$

and the principal stresses are:

$$s_1 = \frac{s_x + s_c}{2} + \sqrt{\left(\frac{s_x - s_c}{2}\right)^2 + s_s^2} \qquad (29)$$

$$s_2 = \frac{s_x + s_c}{2} - \sqrt{\left(\frac{s_x - s_c}{2}\right)^2 + s_s^2} \qquad (30)$$

$$s_3 = s_r = -P_e \qquad (31)$$

The stresses, s_1 and s_2 can be easily established by Mohr's stress circle, *Fig.* 7.

The selected value of t is considered satisfactory if for cast iron drums $s_1 \leqq$ permissible design stress in tension and $s_2 \leqq$ permissible design stress in compression. For mild steel drums the necessary condition is $s_1 - s_2 \leqq$ permissible design stress for mild steel.

In most practical cases of crane design s_a and s_s are very small compared to the other stresses and the previous criteria simplify to: $s_b \leqq$ permissible design stress in tension, $s_c \leqq$ permissible design stress in compression for cast iron and $s_b - s_c \leqq$ permissible design stress for mild steel.

Examples: The procedure outlined and the use of the calculation charts are further illustrated by the following examples. No allowance has been made in the examples for impact effects and wear; one ton = 2000 pounds.

EXAMPLE 1: Design a 27-ton auxiliary hoist drum for a 135-ton electric overhead travelling power house crane.

Hoist Requirements: Height of lift is 100 ft with 8 falls of rope, 2 ropes leading from drum (*Fig.* 2*b*). Rope factor of safety is 6. Drum material is cast iron: $E = 15 \times 10^6$ psi, tensile and shear strength = 35,000 psi, compressive strength = 135,000 psi.

Drum Dimensions: Load in each rope is $T = 28/8 = 3.5$ tons, assuming weight of fall block and rope as 1 ton. Guaranteed strength of rope must not be less than 6 (3.5) = 21 tons. If a 6/37 rope construction of plow steel wire is used, a 2.5-in. circumference rope will be satisfactory and $d = 13/16$ in., $p = 15/16$ in. Drum diameter is not to be less than $19d$; thus $D = 20.188$ in. will be acceptable.

For a 100-ft lift the number of grooves, allowing for anchorage and one spare groove, will be 77. Grooved length at each end of drum = 77 (15)/16 = 72 in. and, allowing 18 in. between the grooved ends, the total length of drum is $l = 162$ in. = 13 ft 6 in.

Drum Thickness: From Equation 1, considering compressive tangential stress, the pressure on the drum is

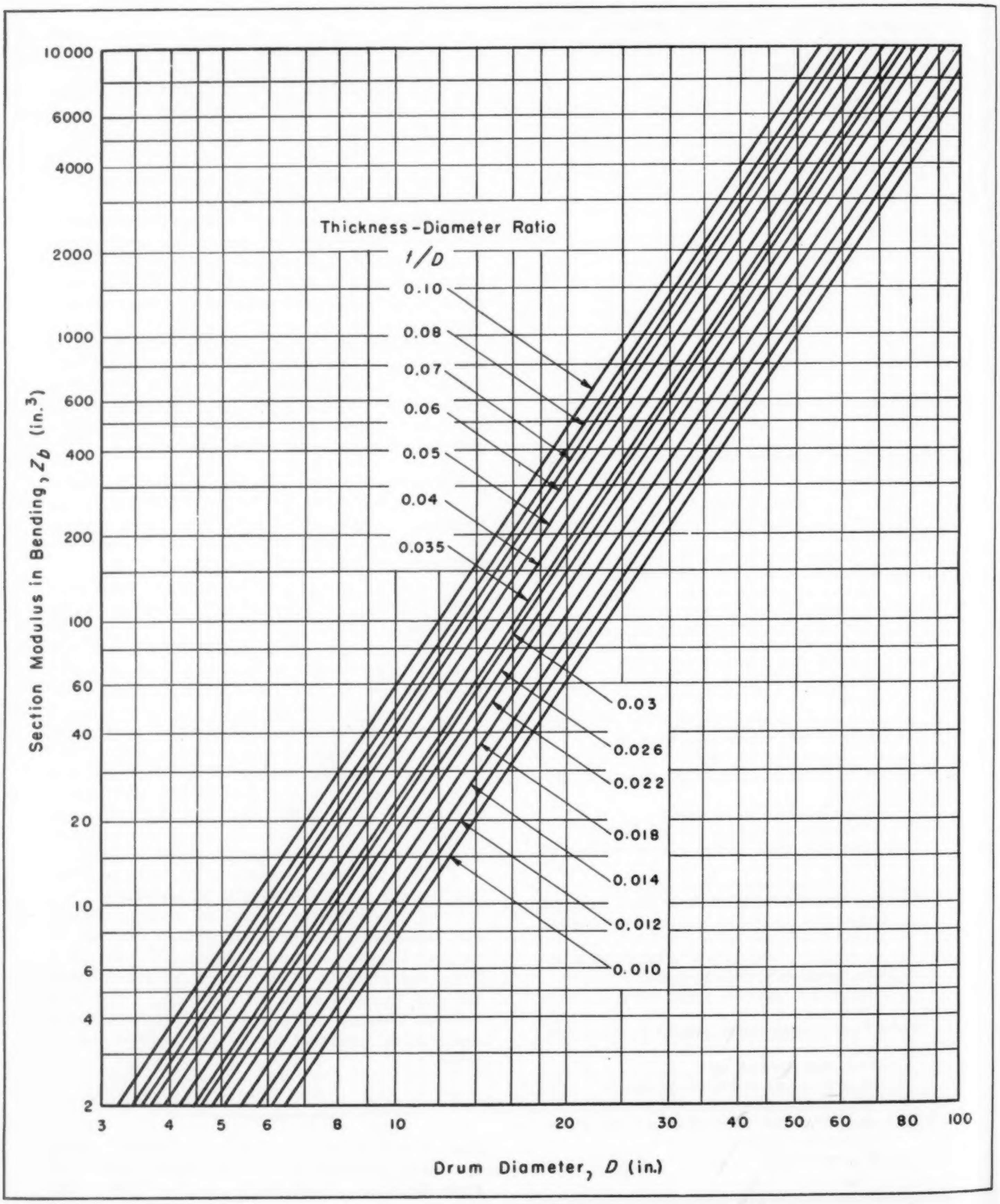
$$P_e = \frac{2(1)(7000)}{0.937(20.188)} = 740 \text{ psi}$$

For a factor of safety of 5, $s_d = 135{,}000/5 = 27{,}000$ psi. From Equation 22, $A = 2\ (27{,}000)/740 = 73$. From *Fig.* 4, $\alpha_1 = 0.015$ and $t = \alpha_1 D + 1/16 = 0.365$

Nomenclature

C, K, k = Constants
d = Diameter of wire rope, inches
D = Outside diameter of rope drum, measured at base of innermost rope layer, inches
E = Young's modulus of elasticity, psi
F = Axial force in multiple-layer drums, pounds
i, o = Subscript notations for inner and outer drum surfaces, respectively
l = Length of drum between end supports, inches
M_b, M_t = Maximum bending moment and maximum torque, respectively, pound-inches
n = Number of rope layers
N, N_c = Service and collapse safety factors, respectively
p = Pitch of rope winding, inches
P_c = Collapsing pressure for drum shell, psi
P_e = External drum pressure from coiling of rope, psi
s_a = Axial stress in drum shell, psi
s_b = Maximum bending stress in drum shell, psi
s_c = Maximum tangential compressive stress in drum shell, psi
s_d = Permissible design stress in compression, psi
s_r = Radial stress in drum shell, psi
s_s = Maximum torsional shear stress in drum shell, psi
s_x = Total normal stress parallel to drum axis, psi
s_1, s_2, s_3 = Principal stresses, psi
t = Wall thickness of drum body, inches
T = Design value of rope tension, pounds
Z_b, Z_t = Section moduli in bending and torsion, respectively, inches^3
μ = Poisson's ratio

Fig. 6—Chart for determining values of section moduli in bending and torsion ($Z_t = 2Z_b$)



in. or preferably, $\frac{3}{8}$ in.

From the criterion for collapse, $l/D = 162/20.188 = 8.04$. With $N_c = 5$, from Equation 18

$$P = \frac{5(740)(10^6)}{15 \times 10^6} = 246.7 \text{ psi}$$

From *Fig.* 5, $\alpha_2 = 0.053$ and thus $t = 0.053\ (20.188) = 1.07$ or $1\frac{1}{8}$ in. In this example, the criterion to resist collapse determines the drum thickness, provided the combined tangential and bending stresses are within allowable limits.

Combined Stresses: From *Fig.* 4 with $t/D = 1.125/20.188 = 0.056$, $A_1 = 19$. From Equation 24, $s_c = 740\ (19)/2 = 7040$ psi. From Equation 4, $M_b = 7000\ (72) = 504{,}000$ lb-in. From *Fig.* 6, $Z_b = 300$ in.^{3} and from Equation 26, $s_b = 504{,}000/300 = 1680$ psi. Since the material is cast iron, the maximum stress theory is applied, giving a factor of safety of $135{,}000/7040 = 19.2$ in compression and $35{,}000/1680 = 20.8$ in tension. This indicates that a lower grade cast iron would be satisfactory for the drum, increasing the economy of the design.

EXAMPLE 2: Design the hoisting drum for a $6\frac{1}{2}$-ton jib crane for cargo handling.

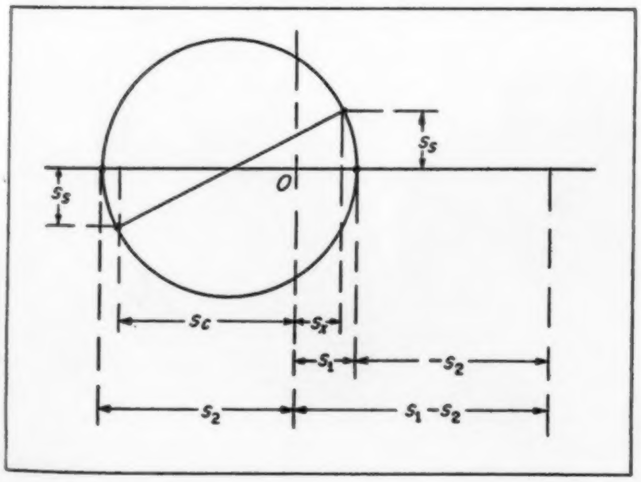
Hoist Requirements: Height of lift is 172 ft with a single fall of rope (*Fig. 2a*). Rope factor of safety is 7. Drum material is cast iron: $E = 12 \times 10^6$ psi, tensile strength = 20,000 psi, compressive strength = 100,000 psi.

Drum Dimensions: Load in the rope is $T = 6.5\ (2000) = 13{,}000$ lb. The rope must have a guaranteed strength of $6.5\ (7) = 45.5$ tons. A 6/37 rope construction of mild plow steel will be used: circumference = $3\frac{7}{8}$ in., $d = 1.25$ in., $p = 1.5$ in. The minimum drum diameter of $19d$ is satisfied by $D = 28.75$ in. The number of grooves for a 172-ft. lift will be 26, allowing 3 grooves for anchorage and one spare groove. Length of drum is $l = 26\ (1.5) = 39$ in.

Drum Thickness: From Equation 1, considering compressive tangential stress and using $N = 5$,

$$P_e = \frac{2(1)(13{,}000)}{1.5(28.75)} = 604 \text{ psi}$$

Fig. 7—Principal stress determination from Mohr's stress circle



Also, $s_d = 100{,}000/5 = 20{,}000$ psi and from Equation 22, $A = 40{,}000/604 = 66$. From *Fig.* 4, $\alpha_1 = 0.0155$ and $t = 0.0155\ (28.75) + 0.063 = \frac{1}{2}$ in. approximately.

From the criterion for safety against collapse, $l/D = 1.36$. From Equation 18, with $N_c = 5$,

$$P = \frac{5(604)(10^6)}{12 \times 10^6} = 252 \text{ psi}$$

From *Fig.* 5, $\alpha_2 = 0.027$, giving $t = 0.027\ (28.75) = 0.775$ in. or preferably, 13/16 in.

Combined Stresses: From *Fig.* 4, with $t/D = 13/16\ (28.75) = 0.028$, $A_1 = 38$. Also, $s_c = 38\ (604)/2 = 11{,}480$ psi. From Equation 3, $M_b = 13{,}000\ (39)/4 = 126{,}800$ lb-in. From *Fig.* 6, $Z_b = 480$ in.^{3} and from Equation 26, $s_b = 126{,}800/480 = 264$ psi. From the maximum stress theory, the factor of safety in compression is $100{,}000/11{,}480 = 8.72$ and in tension $20{,}000/264 = 75.8$.

EXAMPLE 3: Design the main hoist drum on an 11-ton overhead travelling electric crane.

Hoist Requirements: Height of lift is 54 ft with 4 falls of rope, 2 ropes leading from drum (*Fig. 2b*). Rope factor of safety is 7. Drum material is mild steel: $E = 30 \times 10^6$ psi, tensile and compressive strength = 60,000 psi, shear strength = 45,000 psi.

Drum Dimensions: Load in the rope is $T = 11/4 = 2.75$ tons = 5500 lb. Required rope strength is $2.75\ (7) = 19.25$ tons. The rope to be used is of mild plow steel, 6/37 construction, circumference = 2.5 in., $d = 13/16$ in. and $p = \frac{7}{8}$ in. For the drum diameter a value $D = 16$ in. may be used, which is greater than $19d$. Length $l_o = 29\ (7)\ 18 = 25.4$ in. and $l = 25.5 + 9 + 25.5 = 60$ in.

Drum Thickness: From Equation 1

$$P_e = \frac{2(5500)}{0.875(16)} = 785 \text{ psi}$$

Also, with $N = 4$, $s_d = 60{,}000/4 = 15{,}000$ psi. From Equation 22, $A = 2\ (15{,}000)/785 = 38.2$. From *Fig.* 4, $\alpha_1 = 0.028$, which gives $t = 0.028\ (16) + 0.063 = 0.510$ in. or preferably 9/16 in. Also, $l/D = 3.75$ and with $N_c = 4$, from Equation 18

$$P = \frac{4(785)(10^6)}{30 \times 10^6} = 105 \text{ psi}$$

From *Fig.* 5, $\alpha_2 = 0.029$ and $t = 0.029\ (16) = 0.464$ which is less than 0.510, thus $t = 9/16$ in. will be used.

Combined Stresses: From *Fig.* 4, with $t/D = 9/16\ (16) = 0.035$, $A_1 = 30$. Hence, $s_c = -30\ (785)/2 = -11{,}800$ psi, $M_b = 5500\ (25.5) = 140{,}200$ lb-in., $Z_b = 100$ in.^{3} (*Fig.* 6), $s_b = 140{,}200/100 = 1402$ psi. Using the maximum shear stress theory, $s_c - s_b = 1420 + 11{,}800 = 13{,}200$ psi. The factor of safety is $60{,}000/13{,}202 = 4.54$ which is satisfactory.

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Air-Lubricated Bearings

DESIGN ABSTRACTS

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IN RECENT months there has been a sharp focusing of interest on bearings that are lubricated with air. This interest is centered around the two major advantages of the use of air: first, freedom from contamination by leakage of the lubricant, and second, low friction.

Air-lubricated bearings are presently used over wide ranges of speeds, from very low to as high as 80,000 to 100,000 rpm. They have been designed as both journal bearings and thrust bearings and have been lubricated by normal hydrodynamic action or by the hydrostatic introduction of air at elevated pressures. A considerable variety of designs is available to satisfy the needs of almost any type of application, producing low-friction bearings with reasonably good load-carrying capacities. Certain designs have even been used to maintain extremely close running tolerances between machine elements, and one air-lubricated journal bearing has been used to measure pressure differences as small as 0.01-mm of water.

Experimental Work: This all began many years ago when Sir Isaac Newton conducted his great series of experiments. One of these experiments was concerned with the resistance to motion of two concentric cylinders, both being submerged in a liquid, when one was turned relative to the other, *Fig.* 1. Newton observed that for a constant temperature the frictional force F was proportional to the relative speed of rotation v and to the swept area A, and inversely proportional to the radial clearance space between the cylinders h, providing that this clearance space was small. The constant of proportionality depends upon the liquid and is now recognized as the absolute viscosity μ, which gives

$$F = \mu A \frac{v}{h} \qquad (1)$$

As applied to a journal bearing of length l and radius r, A becomes

Fig. 1—Newton's classic experiment on frictional force between two submerged concentric cylinders, the forerunner of rational journal bearing analysis

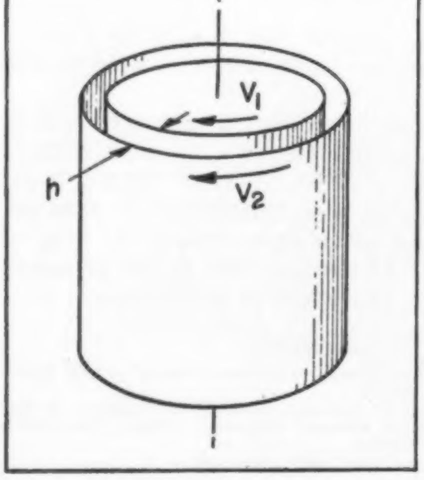
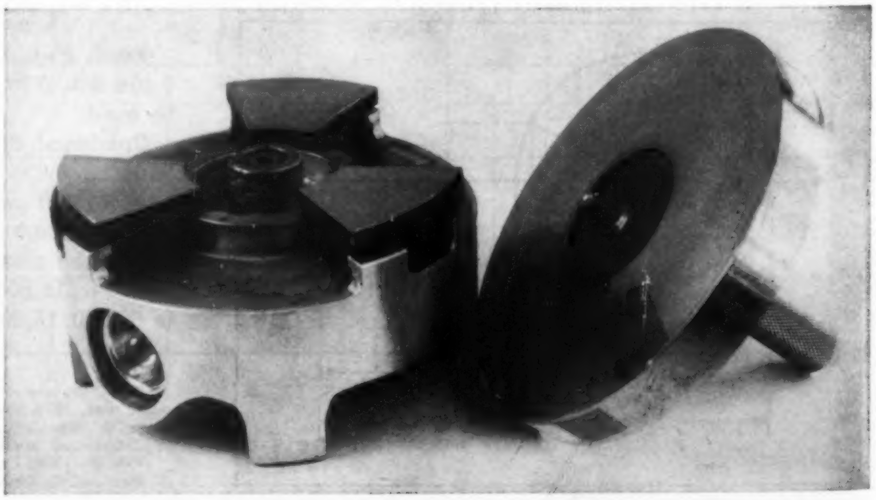


Fig. 2—Model of an air-lubricated Kingsbury thrust bearing which has a coefficient of friction of approximately 0.002



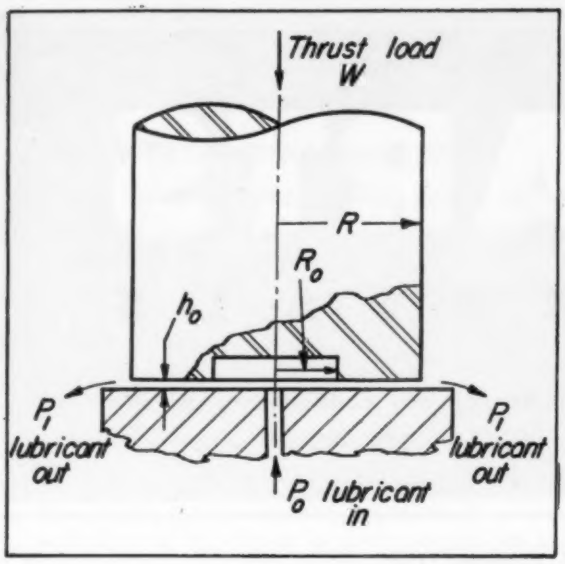


Fig. 3—Left—Elements of a hydrostatic step bearing which maintains a surface separating film even when the bearing is not rotating

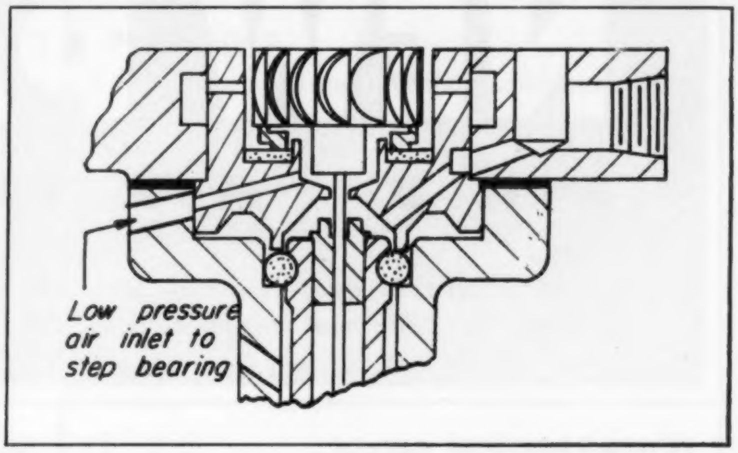
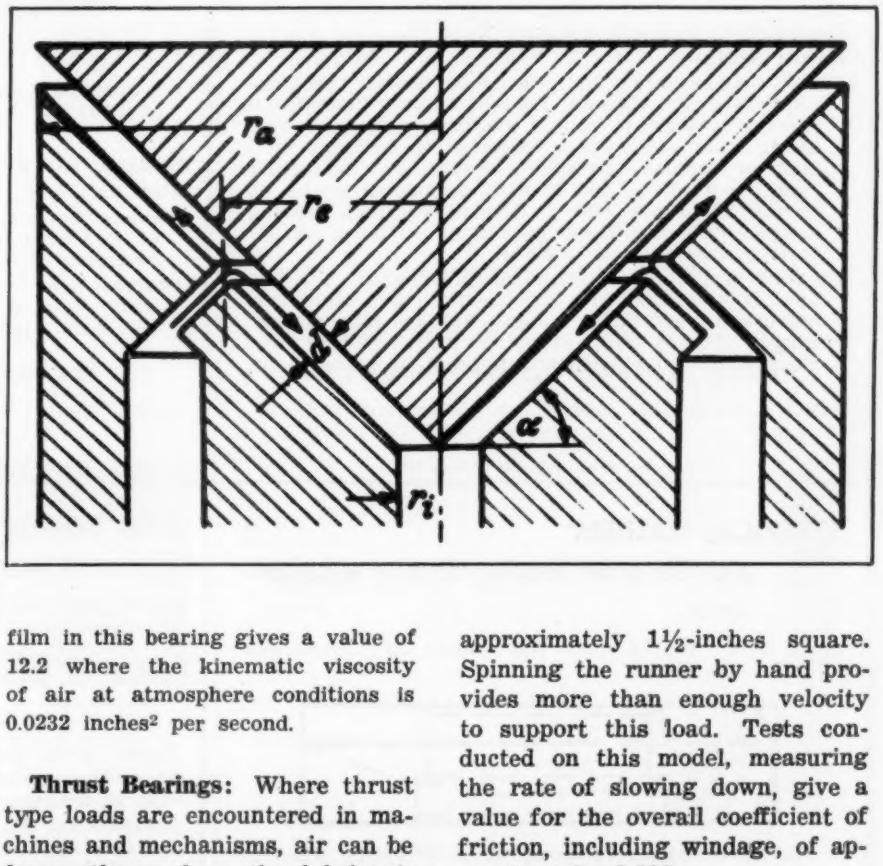


Fig. 4—Above—Ultracentrifuge application of an air-lubricated step bearing which operates at speeds up to 80,000 rpm

Fig. 5—Below—German air bearing design, a modification of the step bearing of *Fig.* 3, in which air flows in two directions



$2\pi rl$, v becomes ωr and the friction torque T is

$$T = \frac{2\pi r^2 l \mu \omega}{h} \quad \ldots\ldots\ldots\ldots\ldots \quad (2)$$

where ω is the angular velocity in radians per second and h is the radial clearance in the bearing.

Albert Kingsbury and others have demonstrated the accuracy of Equation 2 when applied to air-lubricated bearings. Kingsbury experimented with a bearing 6 inches in diameter by 6¼-inches long in which was placed a short length of shaft weighing 50.5 pounds. Radial clearance was about 0.0008-inch. The load was carried without difficulty at shaft speeds above 250 rpm. Kingsbury measured coefficients of friction as low as 0.001. Equation 2 can be used for air lubrication even when the bearing is carrying a moderate load of a few pounds per square inch, as long as a laminar flow condition prevails in the film. This is generally true if Reynolds number, vD/ν, for the film is less than 2000 (ν is the kinematic viscosity of air, v is the surface velocity of the shaft, D is the air film thickness).

Example: A bearing 3 inches in diameter by 4 inches long running in atmospheric air at a speed of 1800 rpm and with a diametral clearance of 0.002-inch would have a friction torque T, calculated from Equation 2, of 0.0279 pound-inch. For this calculation, the viscosity of air at atmospheric conditions is 0.018-centipoise or 1.45 (0.018) (10^{-7}) pounds-second per inch^2 [Reyns]. An evaluation of the Reynolds number, R, for the air film in this bearing gives a value of 12.2 where the kinematic viscosity of air at atmosphere conditions is 0.0232 inches^2 per second.

Thrust Bearings: Where thrust type loads are encountered in machines and mechanisms, air can be frequently used as the lubricant. *Fig.* 2 shows a model of a Kingsbury type thrust bearing that runs very nicely on air. When the runner is turning the pivoted shoes tilt automatically until they develop a wedge-shaped film of air between the shoes and the runner sufficient to carry the applied load. The runner in the model weighs 4 pounds and there are 3 shoes, each approximately 1½-inches square. Spinning the runner by hand provides more than enough velocity to support this load. Tests conducted on this model, measuring the rate of slowing down, give a value for the overall coefficient of friction, including windage, of approximately 0.002.

Where it is desirable to carry the imposed thrust load and separate the bearing surfaces at all times by a film of air, whether there is bearing rotation or not, it is possible to introduce air hydrostatically under pressure. The arrangement of elements in such a bearing, called a "step" bearing, is shown

(*Continued on Page* 381)

NEW PARTS AND MATERIALS

... presented in quick-reference data sheet form for the convenience of the reader. For additional information on these new developments, see page 297

EXTREME-PRESSURE GREASE 1

... operates over wide temperature range

Alpha Corp., 179 Hamilton Ave., Greenwich, Conn.

This grease is an inert compound of molybdenum disulfide and silicone.

Designation: Molykote-Silicone, Type 77.

Form: Grease.

Service: Lubricating under extreme bearing pressure requirements; silicone content makes compound chemically inert to rubber, leather, plastics, Teflon, nylon, fiber and metals; resistant to oxidation and gum or varnish formation; operating temperature range, −50 F to 400 F with occasional peaks up to 600 F; above 600 F silicone volatizes leaving molybdenum disulfide deposited on surfaces for effective lubrication up to 800 F.

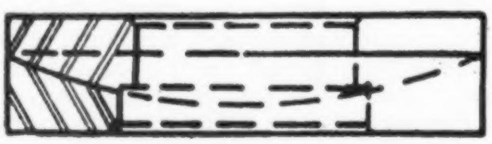
Properties: Grease compound of molybdenum disulfide powder processed with silicone fluid and stabilizer; unworked penetration range, 275 to 325; worked range, 290 to 340; dropping point, testing was stopped at 490 F with no apparent dropping point.

For more data circle MD-1, Page 297

SPHERICAL WASHERS 2

... provide good bearing despite misalignment

Reid Tool Supply Co., 709 Baker St., Muskegon Heights, Mich.



Mating spherical faces of these two-piece washers slide so that full contact is made with both nut and work surface.

Size: For ¼ to 1 in. diam bolts.

Service: Providing good contact for clamping purposes; resist corrosion.

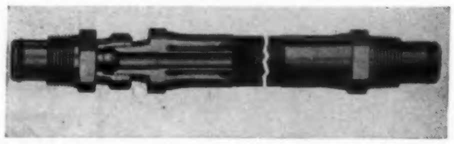
Design: Two-piece washer with spherical mating surfaces; made of high quality low-carbon steel; all sharp edges removed; zinc plated; case hardened.

For more data circle MD-2, Page 297

HOSE CONNECTORS 3

... use O-ring packing

Eastman Mfg. Co., Manitowoc, Wis.



O-ring packing seals against leakage at pressures of 5000 psi though connection is only hand tight.

Designation: MR.

Size: For ½, ¾, 1 and 1¼ in. ID hose.

Service: Connecting wire-braided hose; will not leak at high pressures; O-ring seals easily replaced when necessary; easily connected and disconnected.

Design: Hose connectors; all pipe threads are "Dryseal"; O-ring seals make excessive tightening to withstand high pressures unnecessary.

For more data circle MD-3, Page 297

HIGH DENSITY METAL 4

... 50 per cent denser than lead

Metal Carbides Corp., 107 E. Indianola Ave., Youngstown, Ohio

Can be used for balancing or atomic radiation screening.

Form: Solid.

Size: Hot-pressed parts up to 25 in. OD by 40 in. long; weights in excess of 1000 lb.

Service: For static and dynamic balancing as well as other applications requiring maximum weight in minimum space; resisting radioactive ray penetration.

Properties: Tungsten alloy; produced from powdered metals using a hot-press method; readily machined with carbide tipped tools; specific gravity in excess of 17.5 grams per cu cm.

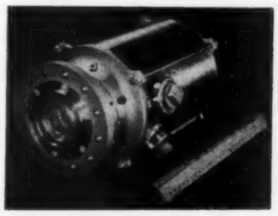
For more data circle MD-4, Page 297

NEW PARTS AND MATERIALS

WATER BRAKE DYNAMOMETER — 5

. . . for high-speed rotary power absorption

Industrial Engineering Co., Island Rd. and Suffolk St., Philadelphia 42, Pa.



Requires no cradle or torque arm since it is mounted directly on the frame of the machine under test.

Designation: Model W200 Hydra-Brake.

Size: 6.63 in. OD, 13.44 in. long; weight, 35 lb.

Service: Absorbing power from any high speed, rotating machine or engine; used in jet engine field for studies of accessory power take-offs; utilizes heat rise-flow principle; operates clockwise or counterclockwise; measures to 600 hp at 8000 rpm; accuracy within 2% tolerance.

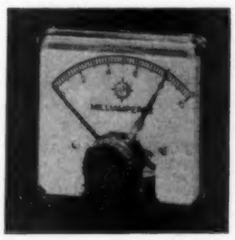
Design: Tortuously-ported stators and disks; self-supporting, end-mounted; drag is caused by whirling disks shearing a layer of water around the inner periphery of brake housing; drag is augmented by acceleration and deceleration of water passing through ports of rotating disks and stators; other models available for power loads up to 1000 hp.

For more data circle MD-5, Page 297

CONTACT METER-RELAY — 6

. . . automatically energizes circuits

Assembly Products Inc., 1002 *Bell St., Chagrin Falls* 1, *O.*



Incorporation of booster contacts in the meter-relay provides an increase in current carrying capacity.

Designation: Simplytrol.

Size: 2½ in. sq, 3¾ in. sq and 4½ in. sq.

Service: Providing sensitive automatic energizing of ac or dc loads when predetermined signal limits are reached; meter ranges, 0-20 to 0-500 for microammeters, 0-1 to 0-1000 for milliammeters, 0-1 to 0-50 for ammeters, 0-1 to 0-500 for voltmeters; booster contact ratings, 1 amp at 28 v or ½ amp at 110 v ac noninductive.

Design: Meter plus relay-locking circuit; stationary pointer on meter is preset at limit value; if a signal causes the moveable pointer to reach stationary pointer position, a meter contact energizes the locking circuit, providing positive locking of a separate pair of load or booster contacts; load-carrying contacts are carried on frame inside meter case; other meter ranges available.

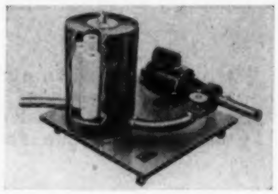
Applications: Sound or light producing alarms; continuous on-off or automatic shut-off switches; bearing temperature warning signals on turbines and generators; automatic speed controls on machines and conveyors.

For more data circle MD-6, Page 297

FILTER PUMP — 7

. . . for acid and alkaline solutions

Sethco, 70-78 *Willoughby St., Brooklyn* 1, *N. Y.*



Filter units, pump, and fittings have corrosion-proof construction.

Designation and Size:

Model	Rated Capacity (gal/hr)	Overall Size (ft)	Weight (lb)
ASI-300	300	2 x 2 x 2	125
ASI-400	400	2 x 2 x 2	135
ASI-600	600	2 x 2 x 2	150

Service: Filtering practically any acid or alkaline solution from pH 1 to pH 14; removes particles down to one micron in size from corrosive solutions; provided with two 12-ft lengths of hose; strainer stops metallic objects, all units furnished with ½-hp, 110 or 220 v ac completely enclosed motor.

Design: Stainless steel volute-type centrifugal pump and fittings; thermosetting ethoxyline resin filter assemblies and strainer; cotton, dynel, Orlon, nylon, spun-glass, porous-stone or porous-carbon filters; phenolic base; acid and alkaline-proof hose.

For more data circle MD-7, Page 297

FLOW RATE METER — 8

. . . measures air and industrial gases

Waukee Engineering Co., 759 *N. Milwaukee St., Milwaukee* 2, *Wis.*

Meters have built-in calibration for the specific gas and rate of flow to be measured.

Designation and Size:

No.	Length (in.)	Diam (in.)	Pipe Connection (NPT, in.)
M1 to M7	19¼	2 1/16	¾
M8 to M10	20⅛	2 5/16	1¼

Service: Measuring flow rates of air and industrial gases including ammonia, dissociated ammonia, propane, butane, city gas, endothermic and exothermic gas, hydrogen, natural gas, nitrogen, and oxygen; flow-rate capacities available for most gases to 1500 cu ft per hr; specifications required when ordering —gas to be measured, gas specific gravity, gas temperature, gas pressure, max flow, desired meter size, and desired name tab.

Design: Float assembly in tapered metering tube; float assembly consists of float, float rod, and indicator; indicator in oil-filled pyrex tube; calibrated 6 in. scale.

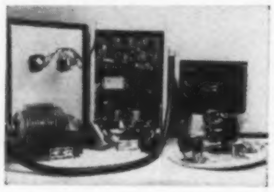
For more data circle MD-8, Page 297

NEW PARTS AND MATERIALS

ELECTRONIC GOVERNORS 9

. . . maintain speed within one-half per cent

W. C. Robinette Co., 802 Fair Oaks Ave., South Pasadena, Calif.



May be applied to almost any prime mover and will maintain speed over the entire torque range.

Designation: Motron Servo.

Service: Maintaining prime mover speed; will not overshoot or hunt when correcting speed; starts to correct when speed is in error by 0.02%; may be provided to average out errors over a period of time.

Design: Small centrifugal governor is driven by prime mover through epicyclic differential gear; other input to differential is two-phase induction servomotor; input to servomotor is from power line and from amplifier controlled by governor; variation of prime mover speed causes governor to vary servomotor input, causing the servomotor to reset the throttle or other speed-controlling device; system is specially designed for application using standard package servo to give quick working mockup.

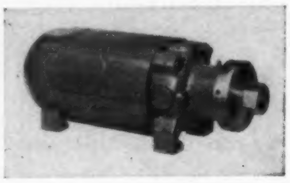
For more data circle MD-9, Page 297

ARBOR MOTORS 11

. . . for tools mounted directly on shafts

Louis Allis Co., 427 E. Stewart, Milwaukee 7, Wis.

Extremely rigid construction permits mounting of saws, grinding wheels, cutter heads, etc. directly on the shafts of these motors.



Designation and Size: Series 600—3, 5 and 7.5 hp, 3¼ in. from center line of shaft to flat side of motor; Series 800—7.5, 10 and 15 hp; 4¼ from centerline of shaft to flat side of motor.

Service: For continuous operation at rated hp at 3600 rpm; rated 55 C rise; high breakdown torque; smooth running.

Design: Totally enclosed, fan-cooled electric motors; mounting feet located on end bells inside maximum width of end bells; shaft extension end of shaft supported by single-row, double-row or duplex bearing, depending on service; connector for single-voltage leads or conduit box for dual-voltage leads may be opposite feet or on either side of motor as required.

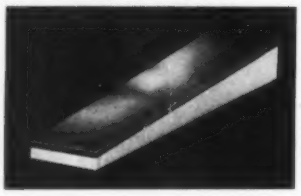
For more data circle MD-11, Page 297

WEAR STRIPS 10

. . . have aluminum-bronze wearing surface

Ohio Knife Co., Dreman Ave., Cincinnati 23, O.

Aluminum-bronze wearing surface is claimed to be superior to conventional cast phosphorous bronze.



Size: 72 in. lengths; ¾, 1 or 1¼ in. thicknesses; 1½, 1¾, 2, 2¼, 2½, 3, 4, 5, 6 and 8 in. widths.

Service: Providing long-wearing bearing surfaces; backing material easily tapped, drilled or otherwise machined.

Design: Strips of soft machine steel with welded-on layer of Ampco aluminum bronze; aluminum bronze layer is finish-ground, flat and parallel; other sizes available as specials.

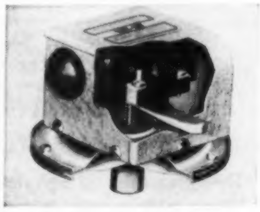
For more data circle MD-10, Page 297

VACUUM-OPERATED SWITCH 12

. . . breaks circuit when vacuum is low

Jaycon Associates, 404 N. Washington Ave., Minneapolis 1, Minn.

Will stop gasoline engines or electric motors in pumping service when liquid source runs dry.



Designation: Vac-On 35-2 and 35-5.

Size: 4¼ in. diam, 3½ high; ¼-in. NPT female vacuum connection.

Service: Stops electric motors to 2 hp on ac, to ½-hp on dc, or on any spark-ignition engine when vacuum applied to switch is less than 1½ in. of Hg; for heavy-duty service.

Design: Diaphragm-operated switch; contact points normally closed; lever incorporated for reclosing of points after opening; contact points are silver.

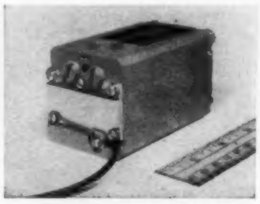
For more data circle MD-12, Page 297

NEW PARTS AND MATERIALS

SERVO CONTROL VALVE 13

. . . controls pressure with small current

Standard Controls Inc., 1230 Poplar Place, Seattle 44, Wash.



Eight milliampere differential current flow regulates 3000 psi hydraulic output.

Designation: PC-1.

Size: 2 by 2 by 3½ in.; weighs 1.4 lb; ports are ⅝-in. tube size per AND 10050-6.

Service: Controlling hydraulic motor or actuator positioning devices; for supply pressures from 500 to 3000 psi; proof pressure of pressure section is 4500 psi, of return section 1500 psi; burst pressure, 7500 psi minimum for pressure section, 3000 psi minimum for return section, coil resistances, 700 ohm; rated 4.2 hp maximum; operates from −65 to 250 F; no measurable effect on pressure regulation with 10*g* acceleration along any axis.

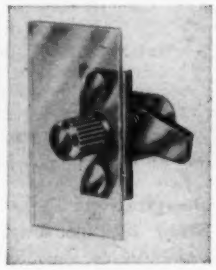
Design: Electrically controlled pressure-control valve; two coils connected in opposition control miniature electric torquer which operates a pair of push-pull slide valves to regulate output pressures; 8 in. 4-wire color-coded cable for electrical connections; output of valve may drive motor or actuator in either direction.

For more data circle MD-13, Page 297

ADJUSTABLE PAWL LATCHES 15

. . . for small doors and panels

Southco Div., Southchester Corp., Lester, Pa.



Though small, high strength is claimed for this latch.

Designation: No. 43.

Size: 1⅛ in. long by ½-in. wide by 7/64-in. deep housing; extends 0.859-in. back of housing; total depth is 1.359 in.; operating knob approximately ⅜-in. in diam.

Service: Retaining panels or doors in material from ⅛ to ½-in. thick; resist vibration; easily adjusted for materials of varying thicknesses; latching pressure may be varied to compress gasket; easily installed by riveting or spot welding.

Design: Knurled operating knob turns spring-retained pawl on threaded shaft; additional turns of knob cause pawl to move toward or away from panel to increase pressure or increase grip; operating knob is brightly plated; slotted-head knobs available.

For more data circle MD-15, Page 297

HIGH-PRECISION POTENTIOMETERS 14

. . . in new smaller sizes

Helipot Corp., South Pasadena, Calif.

Electrically equivalent to the previous Model F, these potentiometers are 1/16-in. smaller in OD.

Designation: Model L.

Size: 3 in. diam; length varies with number of ganged sections and is 4.912 in. for 8 sections.

Service: Continuously varying electrical resistance in as many as 8 circuits simultaneously; resistance per section of 5000 to 100,000 ohm; linearity of ±0.5% standard; linearity as high as ±0.1% on special order; phasing of sections may be varied.

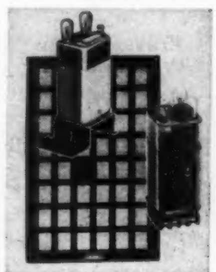
Design: Precision-ganged potentiometers; all models have phasing clamps; single section may have as many as 33 taps; extra taps are accurately spot-welded at the factory; shaft rotation is continuous; all electrical connections are spotwelded; models for bushing or servo-lid mounting with sleeve, Oilite or ball bearings available.

For more data circle MD-14, Page 297

ALARM RELAYS 16

. . . incorporate warning lights

Tigerman Engineering Co., 4332 N. Western Ave., Chicago 18, Ill.



Completely enclosed and dustproof, these units may be readily disassembled if necessary.

Designation: LPG.

Service: Operating warning devices such as bells or horns in addition to lighting or flashing of integral light or lights; easily connected; contacts rated 2 amp at 125 v 60 cycle ac, noninductive load.

Design: Relays with normally-open or normally-closed maintained contacts, momentary contacts, or normally-open or normally-closed lock-in contacts; 10 different contact combinations available; also available in multiple unit systems in cabinets with wired chassis; engraved glass plates for identification of circuits in cabinets or colored lenses are available.

For more data circle MD-16, Page 297

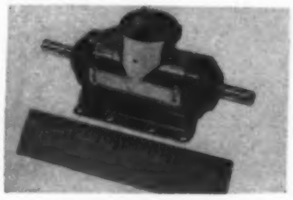
NEW PARTS AND MATERIALS

MINIATURE SPEED CHANGER 17

. . . has lever adjustment

Metron Instrument Co., 432 Lincoln St., Denver 9, Colo.

Changer can be controlled locally or remotely with connecting links, cams, cranks, or similar mechanisms.



Designation: Type 3B.

Size: 4 5/32 in. long, 1½ in. wide, 2 1/16 in. high; lever radius, 1 in.; lever arc, 63 deg; weight, 6 oz.

Service: Providing variable speed-changing over 25 to 1 range; lever orients independently of pointer to any horizontal position; lever can be removed, giving a vertical speed control shaft; remote ratio indication can be arranged by a scale associated with the remote speed regulator; settings can be made while running or stationary; operates in any position.

Design: Input and output drive disks make contact through two rollers pivoted to contact disks at changeable radii; permanently lubricated ball bearings; completely enclosed.

Applications: Chart drives, instrument test equipment, mixers, optical drives, signal generators.

For more data circle MD-17, Page 297

ELECTRONIC TIMER 18

. . . flexible, accurate and dependable

Photoswitch Inc., 77 Broadway, Cambridge 42, Mass.

Provides four fundamental types of timing.

Designation: 30HL1.

Service: For applications requiring long life, repeat-cycle operation or precise accuracy of timing; provides interval, delayed action, automatic repeat and program timing with variations on each; these timing combinations incorporated in one timer and may be utilized by changing external connections to terminal board; time intervals, 1/20 second to 4 minutes; basic electronic circuit self-compensates for line voltage changes; accuracy, 2%; power requirement—115 or 230 v, 50-60 cycles, 25 w; main contact rating—10 amp, 115 v ac noninductive or 5 amp, 230 v ac noninductive; auxiliary contact rating—1 amp, 115 v ac noninductive or 0.5-amp, 230 v ac noninductive.

Design: Electronic timer employing one electron tube; relay—two single-pole, double-throw switches.

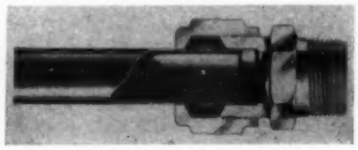
For more data circle MD-18, Page 297

TUBE FITTINGS 19

. . . require no flaring of tubes

Chicago Forging & Mfg. Co., 2000 Southport Ave., Chicago 14, Ill.

Fittings are shock absorbing and vibration-fatigue proof.



Designation: Sealastic.

Form and Size: Seal rings, ferrules, nuts, male connectors, female connectors, unions, 90-deg union elbows, 90-deg elbows, 45-deg elbows, tees and crosses for ⅛ to 1½ in. OD tubes.

Service: Providing fittings for metal tubing on fuel, oil, water, refrigerant, hydraulic, air, and injection lines; finger-tightened fittings withstand pressures up to 4000 psi; when wrench tightened, will sustain pressure resistance beyond that of the tube; handle temperatures from −70 F to 400 F; exact alignment of tubes not necessary for coupling.

Design: Connector nipple, synthetic-rubber seal fixed to tube by a metal ferrule and compression nut; sufficient tube clearance is provided in connection and nut to insure against metal contact between tube and fitting.

For more data circle MD-19, Page 297

OIL-FOG LUBRICATOR 20

. . . produces fine particles at low air flow

C. A. Norgren Co., 3400 S. Elati St., Englewood, Colo.

Assures delivery of adequate oil fog to all lubrication points with a minimum air flow.

Designation and Size:

No.	Inlet Diam (in.)	Height (in.)	Diam (in.)	Oil Capacity
30-41-2S	¼	6 [illegible]	3⅝	½ pt
X3400-2S	¼	14 [illegible]	7	1¾ gal
Y3400-2S	¼	16⅝	10	4 gal

Service: Lubricating and cooling bearings, spindles, gearboxes and small air-powered devices; assures uniform rate of oil feed by constant oil level; 360 deg visibility of oil flow; full view of oil supply; rated air requirements, 0.8-cfm at 10 psi to 6 cfm at 80 psi; safe operating pressure, 150 psi max; safe temperature, 120 F max.

Design: Controlled air-flow oil sprayer; transparent dome-type sight feed; transparent oil reservoir; straight-through pipe connections.

For more data circle MD-20, Page 297

NEW PARTS AND MATERIALS

REMOTE CONTROL TURNTABLE 21

. . . variable speed, reversible unit

Gale Dorothea Mechanisms, 81-01 Broadway, Elmhurst, L. I., N. Y.

Speed of rotation can be selected by turning a dial in a remote control box.

Designation: Flexi-Turn.

Size: 6 in. high, 20 x 24 in. base plate.

Service: Providing variable speed reversible motion; reverses instantly; speed variable from 0 to 20 rpm; accelerates quickly or slowly; reversing and speed switches can be modified for foot operation to free operator's hands; load carrying capacity, 200 lb; power requirement, 115 v 60 cycles.

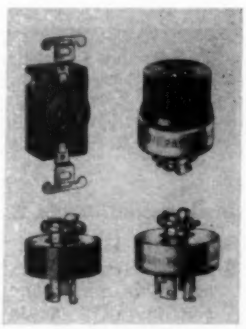
Design: Turntable consists of 11 in. diam iron casting supporting 18 in. diam plywood disk driven through speed reducer by 1/15-hp motor; remote control consists of on-off switch, signal which indicates when table is turning, speed control, forward-reverse switch, and a switch to double or halve the speed instantly; modified controls to fit job needs are available.

For more data circle MD-21, Page 297

WIRING DEVICES 22

. . . quickly and easily locked in place

Pass & Seymour Inc., Solvay Station, Syracuse 9, N. Y.



Caps turn in connections and receptacles to provide positive locking.

Designation: Turnlok.

Service: Providing locked electrical connections; 2-wire caps, connectors, and single flush receptacles with 20 amp, 250 v ac ratings; 3-wire caps, connectors, and single or double flush receptacles with 15 amp, 125 v and 10 amp, 250 v ac ratings; 3 and 4-wire caps, connectors, and single flush receptacles with 20 amp, 250 v and 10 amp, 600 v ac ratings.

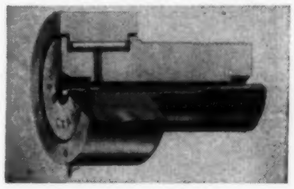
Design: Cord-grip caps and connectors; nonpolarized 2-wire caps; polarized 2, 3 and 4-wire caps, connectors and receptacles; available in rubber and black plastic.

For more data circle MD-22, Page 297

CAM FOLLOWER ROLLER 23

. . . for heavy duty requirements

Smith Bearing Co., 18 Bear Tavern Rd., W. Trenton, N. J.



Stud is large in diameter in comparison to roller size.

Designation: Type CTA.

Size: Roller, stud and thread sizes to specification.

Service: Following cam contours; used in high speed and heavy load applications where unusual strength is demanded; integral flange on stud allows large stud diameter.

Design: Heavy duty cam follower roller; stud—material, 4615 steel or equivalent in carburizing grade, bearing end case-hardened to 58-60 Rockwell C, thread end left soft, SAE fine threads, provision for drive-type grease fitting for lubrication, fitting not provided; outer roller—material, 52100 steel, through-hardened to 60-62 Rockwell C; needle rollers—material, 52100 steel, through-hardened to 58-60 Rockwell C; snap ring—material, 4130 steel or equivalent, hardened and drawn to spring temper, cadmium plated; bearings are prelubricated.

For more data circle MD-23, Page 297

HARD RUBBER COMPOUND 24

. . . for high temperature applications

American Hard Rubber Co., 93 Worth St., New York 13, N. Y.

This compression-molded thermosetting material is available in four forms.

Designation: Tempron.

Form and Size: Pipe—1 to 8 in. diam, 10 ft lengths; fittings—companion flanges for 1 to 6 in. pipe, couplings for 1 to 4 in. pipe, 45 deg elbows for 1 to 3 in. pipe, 90-deg elbows for 1 to 6 in. pipe, tees for 1 to 6 in. pipe; rods, 1/16 to 1/8-in. OD in 30 in. lengths, 5/32 to 3¼ in. OD in 42 in. lengths, 3¼ to 6 in. OD in 30 in. lengths; tubes, 1/16 to 3 in. ID in 30 or 42 in. lengths; sheets, 24 x 50 in., 24 x 25 in., 24 x 12½ in. in 1/64 to 4 in. thickness; molded parts; hand-built sheet-formed parts.

Service: Providing rigid chemically-resistant, lightweight pipe and fittings, rods, tubes, sheets, molded and sheet-formed parts; can be machined and polished; can use in electrical parts; inserts can be molded in or inserted after molding; resists organic and inorganic corrosives and chemicals.

Properties: Based on nitrile synthetic rubber (Buna-N); tensile strength, 7170 to 7500 psi; flexural strength, 11,300 to 11,800 psi; elongation, 2.7 to 2.8%; specific gravity, 1.24 to 1.25; hardness—Scleroscope from 75 to 80, Durometer D 87, Rockwell P from 106 to 115, Rockwell R from 85 to 87; heat distortion, 260 to 275 F.

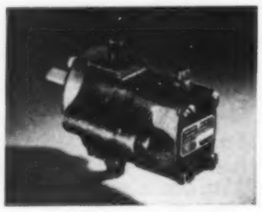
For more data circle MD-24, Page 297

NEW PARTS AND MATERIALS

GEAR PUMPS 25

. . . for No. 2, 3, 4, or heated No. 5 fuel oil

Webster Electric Co., 1900 Clark St., Racine, Wis.



Also available as 115/230 volt pump-motor combination units.

Designation: LAM, LAMV.

Size and Service: 2⅛ in. wide, 2⅞ in. high; ¼-in. pipe inlet and outlet ports;

Type No.	Overall Length inc. shaft ext. (in.)	Capacity* at 100 psi (gph)	Type No.	Overall Length inc. shaft ext. (in.)	Capacity* at 100 psi (gph)
1 LAM	2	10	1 LAMV	2[illegible]	10
2 LAM	2 3/16	25	2 LAMV	2[illegible]	25
3 LAM	2[illegible]	50	3 LAMV	3[illegible]	50
4 LAM	2[illegible]	57	4 LAMV	3[illegible]	57
5 LAM	2[illegible]	65	5 LAMV	3[illegible]	65

* Using No. 2 fuel oil at room temperature at 1750 rpm.

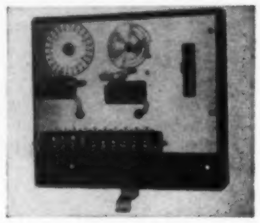
Design: Helical gear pumps; body of cast iron; other parts of selected alloy steels; internal relief valves used in LAMV pumps for regulation between 50 and 200 psi; shaft dimensions and types optional; port sizes may be altered and port location changed under certain circumstances.

For more data circle MD-25, Page 297

REPEAT-CYCLE TIMERS 27

. . . control both on and off time

Paragon Electric Co., Two Rivers, Wis.



Any set schedule of on and off-time is continuously repeated.

Designation: CR3MB-KT2.

Size: 10¼ by 9 by 4½ in.; weighs 10 lb.

Service: For accurate, adjustable, short-duty operations at relatively infrequent intervals; regulates time at which load circuit contacts are closed and length of time during which contacts remain closed; load contacts rated 10 amp at 125 v ac, 5 amp at 240 v ac; for 24 v, 50 or 60 cycle, 120 v, 25, 50 or 60 cycle or 240 v, 25, 50 or 60 cycle operation; on-time ranges of zero to 2½, 5, 7½, 10, 15 or 30 min, off-time ranges of 5 to 60 min, 10 min to 2 hr, 20 min to 4 hr, 30 min to 6 hr, 30 min to 12 hr or 1 to 24 hr.

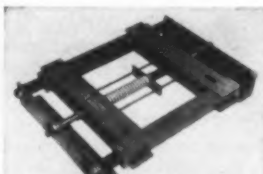
Design: Two interconnected synchronous motor operated single-pole, double-throw switches; switches actuated by pins (off-time) or cam (on-time) on dials mounted on motor shafts; motors are permanently lubricated; entire unit contained in heavy steel case finished in baked enamel.

For more data circle MD-27, Page 297

SLIDING MOTOR BASE 26

. . . automatically maintains belt tension

Automatic Motor Base Co., Windsor, N. J.



Automatic compensation for a predetermined amount of drive belt stretch minimizes maintenance.

Designation: Series BB.

Size: For motors in NEMA frame sizes from No. 364 through 505.

Service: Automatically moves electric motors up to 6½ to 8 in. to maintain belt tension regardless of motor weight, reactive torque or belt angularity; maintains motor alignment and absorbs shocks; either direction of pulley rotation may be used; motor and base may be mounted in any position.

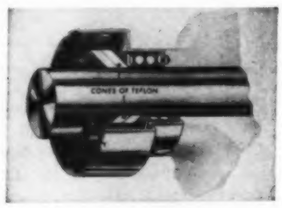
Design: Motor mounting frame is moved by spring in compression to take up slack and maintain belt tension; frame moves on linear ball bearings, inner races are cylindrical tubes of high-carbon steel, outer races are square tubular members of frame; bearing assemblies are grease-packed for life and sealed; thrust bearing facilitates initial adjustment of belt tension.

For more data circle MD-26, Page 297

SHAFT SEAL 28

. . . has Teflon sealing cones

Crane Packing Co., 1800 Cuyler Ave., Chicago 13, Ill.



Teflon sealing cones make this seal usable from temperatures of −100 to 450 F.

Designation: Type 19.

Size: For ¼, ⅜, ½, ⅝ and ¾-in. diam shafts.

Service: Sealing shafts against leakage of oil, water or acids and other corrosive substances; has long service life; seals to 200 psi pressure; for high shaft speeds.

Design: Retainer shell of bronze, stainless steel or other metal as required; carbon seal with lapped face and spring-loaded sealing cones of Du Pont Teflon; has drive ribs for rotating service or can be made in press-in type for stationary service.

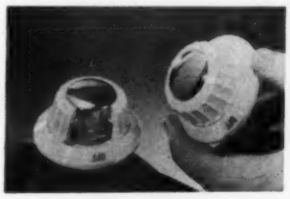
For more data circle MD-28, Page 297

NEW PARTS AND MATERIALS

SAFETY KNOBS 29

. . . cannot be unintentionally turned

Patent Button Co. of Tennessee Inc., Plastic Molding Div., Knoxville, Tenn.



Teeth on operating portion of knob must be engaged with teeth on portion on shaft before shaft can be turned.

Designation: Spin Free.

Size: Approximately 2⅛ in. diam by 1 in.; for standard American Gas Association D-shaped split shafts approximately $\frac{11}{32}$-in. in diam.

Service: Preventing accidental turning of gas valve or electrical switch shafts; tests indicate life in excess of 115,000 operations; knobs resist burning, scratching, grease, oil and are nonelectrostatic.

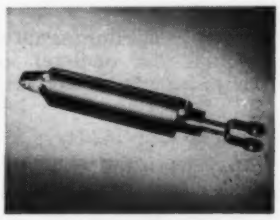
Design: Two-piece knob molded of Plaskon 3555; operating knob must be pushed in to engage portion on shaft before shaft can be turned; light spring holds two pieces apart unless pressure is applied; also available for other sizes and types of shafts.

For more data circle MD-29, Page 297

PNEUMATIC CYLINDERS 31

. . . are easily serviced

Carter Controls Inc., 2800 Bernice Rd., Lansing, Ill.



Easily removable heads and unit rod-end bearing and packing facilitate disassembly for any necessary servicing.

Size: From 1⅜ in. OD and 1⅛ in. bore diam to 8½ in. OD and 8 in. bore diam; stroke lengths to 22 ft; pipe ports from ¼ to 1 in., varying with bore.

Service: Rated pressure, 150 psi; easily disassembled and serviced; long-wearing; piping connections easily made; safety factor of 10 to 1 in all sizes.

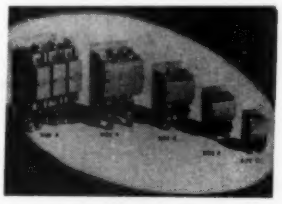
Design: Cylinders of heavy wall tubing; piston-rod scraper ring prevents abrasive material from entering cylinder; heads locked in counterbored section of cylinder by half-hard brass keys, and may be rotated through 360 degrees; keys inserted through milled slots in OD of cylinder; rod-end bearing and seals are an easily removable assembly, retained by internal snap ring; all standard mounting arrangements available.

For more data circle MD-31, Page 297

MAGNETIC MOTOR STARTER 30

. . . is small and lightweight

Arrow-Hart & Hegeman Electric Co., Hartford 6, Conn.



Equipped with vertical overload relays below the contactor.

Designation: Type RA-V.

Size:

NEMA Size (No.)	Height (in.)	Width (in.)	Thickness (in.)
0 and 1	6½	3½	3⅜
2	8	4½	4½
3	10¾	5¾	5⅝
4	14½	7½	7
5	16½	9¾	8⅜

Service: Used where motor starters of reduced size and weight are needed; can be used with Crouse-Hinds M52 type EPC explosionproof condulet.

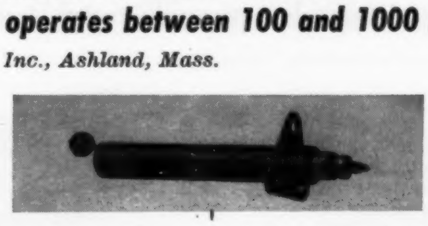
Design: Straight-through front wiring; copper contacts with silver-alloy tips; alkyd tongue and groove construction hood and base; bimetallic overload relays with thermal inverse time-delay snap action; reversing and two speed models also available.

For more data circle MD-30, Page 297

ADJUSTABLE OVERHEAT DETECTOR 32

. . . operates between 100 and 1000 F

Fenwal Inc., Ashland, Mass.



Provides dependable overheat detection in high ambient temperatures produced by aircraft engines.

Designation: Model 17343-29, Type E-4.

Size: 5 in. long, ⅝ in. diam; weight, 0.135-lb.

Service: For aircraft fire and overheat detection; capable of prolonged operation from 100 F to 1000 F; will withstand exposure to −100 F temperatures indefinitely and high accelerations without appreciable effect on dependability; unit is repeatable after exposure to 2000 F flame—a safety factor in recurring fires; response mechanism is not affected by dirt, moisture, or other contaminants; electrical contacts of detector close when ambient temperature rises to preset actuation temperature; rating, 3 amp at 28 v dc.

Design: Temperature-sensitive contact assembly housed in stainless steel shell; hermetically sealed.

For more data circle MD-32, Page 297

NEW PARTS AND MATERIALS

PHENOLIC MOLDING COMPOUND 33

. . . has high-impact strength

Durez Plastics and Chemicals Inc., North Tonawanda, N. Y.

Glass fiber impregnated with phenolic molding compound for one-step compression moldings having high-impact strength.

Designation: 16221.

Form: Dry, impregnated glass-fiber rovings.

Size: 1 in. lengths, other lengths special.

Service: Molded by standard compression molding methods at 5000 to 6000 psi and temperatures of 300 to 350 F; cure time slightly slower than for general purpose materials; breathing cycles recommended.

Properties: Distorted by heat above 450 F;

Specific gravity	1.97	Step-by-step dielectric strength (v/mil, min)	200
Water absorption (% in 24 hr max)	1.00	Dissipation factor, ave	
Impact strength (lb-ft/in., min)	10	at 60 cycles per second	0.03
Flexural strength (psi, min)	18,000	at 1 kilocycle per second	0.02
Tensile strength (psi, min)	6,000	at 1 megacycle per second	0.02
Compressive strength, perpendicular to fibre (psi ave)	17,000	Dielectric constant, ave	
Compressive strength, parallel to fibre (psi, ave)	26,000	at 60 cycles per second	7.0
Modulus of elasticity in tension (psi)	2.4×10^6	at 1 kilocycle per second	6.8
Short time dielectric strength (v/mil, min)	250	at 1 megacycle per second	6.5
		Arc resistance (seconds, ave)	60

For more data circle MD-33, Page 297

IDENTIFICATION PLATES 35

. . . adhesive-backed for easy attachment

Metalcraft Inc., Mason City, Iowa

Resilient, solvent-reactivated adhesive permanently adheres to any grease-free service.

Designation: Numbergraphs and Serially Numbered Autographs.

Size: Numbergraphs are 0.008-in. thick and 1 in. long by 3/10 in. wide or 2 in. long by 0.4-in. wide; Autographs are 0.016-in. thick and 1¾ in. long by $\frac{7}{16}$-in. wide or 2⅜ in. long by $\frac{9}{16}$-in. wide.

Service: Quickly, easily and permanently attached; adaptable to almost any product or surface; Numbergraphs easily shaped to conform to irregular surfaces.

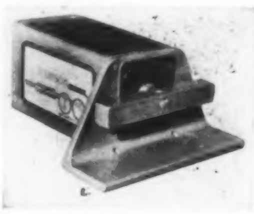
Design: Numbergraphs of mill-finish aluminum; Autographs of chromed metal; numbers $\frac{3}{32}$ or ¼-in. high deeply stamped in metal; Numbergraphs bear serial numbers only; Autographs bear serial numbers, lithographed company name and short advertising message, if desired, in two colors; solvent for reactivation of adhesive backing and brushes for application supplied with plates.

For more data circle MD-35, Page 297

FOOT-OPERATED SWITCH 34

. . . withstands extremely rough treatment

Aronson Machine Co., Arcade, N. Y.



An indestructible, oiltight and dusttight steel enclosure assures long life for this switch.

Designation: Model FPC.

Size: 7 in. long, 6 in. wide and 3 in. high; ½-in. pipe threaded opening for connections.

Service: Starting, stopping and reversing electric motors rated ¾-hp at 110 or 220 v ac single phase, or 110, 220, 440, 550 v ac 2 or 3-phase, or ⅛-hp dc motors operating on 115 or 230 v; breaks all lines to motor in 2 or 3-phase operation and both sides of armature and field circuit in dc operation; unaffected by dirt or dust.

Design: Reversing, foot-operated toggle switch; consists of Allen-Bradley Bulletin 350 Size 00 drum switch in heavy steel case; front of case is 3 by 2 by ¼-in. L, back is ¼-in. plate, remainder of enclosure formed from ⅛-in. plate; operating treadle is ⅝-in. square bar; enclosure is gasketed; switch may also be operated by hand.

For more data circle MD-34, Page 297

CENTRIFUGAL PUMP-MOTOR 36

. . . single unit, highly compact

Goulds Pumps Inc., Seneca Falls, N. Y.

Pump impeller is threaded directly to motor shaft to form an extremely compact unit.

Designation: Fig. 3642.

Size: From 13⅝ in. long, 9⅞ in. high and $8\frac{13}{16}$ in. wide to $16\frac{13}{16}$ in. long, 9⅞ in. high and $8\frac{13}{16}$ in. wide; weigh from 52 to 68 lb; 1 in. or 1¼ in. NPT suction ports; 1 in. or 1¼ in. NPT discharge ports.

Service: Maximum capacity of smallest size is 58 gpm at 30 ft head at 3500 rpm with ¼-hp motor; maximum capacity of largest size is 110 gpm at 40 ft head at 3500 rpm with 2 hp motor; will not leak; easily serviced; working pressure 100 psi, test pressure 150 psi.

Design: Centrifugal pumps mounted directly on electric motors; pumps available of all iron construction, bronze-fitted iron construction or all-bronze construction; casing is volute type; mechanical shaft seal; statically balanced impellers; ball bearings in motor; units are foot mounted; pump housing may be mounted in 8 different positions, 45 degrees apart, to facilitate piping.

For more data circle MD-36, Page 297

NEW PARTS AND MATERIALS

TIME TOTALIZER 37

. . . small and highly accurate

R. W. Cramer Co., Centerbrook 5, Conn.

Accuracy of this $3\frac{3}{4}$ in. diam stopclock is 0.02 of 1% or better.

Designation: Model ET.

Size: $3\frac{3}{4}$ in. diam, extends 1 in. in front of panel; ac models extend 4 in. back of panel; dc model without radio interference filter extends $4\frac{11}{32}$ in. back of panel; dc model with filter extends $4\frac{11}{16}$ in. back of panel.

Service: Indicating time intervals to 60 seconds or to 60 minutes; for 110, 220 or 440 v 50-60 cycle operation, 115 v 400 cycle ac operation, or 28 v dc; 400 cycle ac model also requires 28 v dc for stopping and starting; withstands 30*g* shock and 10 to 55 cycle per second vibration with 0.06-in. excursion; commercial model operates at 0 to 125 F, military model operates from −55 to 85 C; operates at altitudes to 50,000 ft.

Design: Synchronous motor driven stopclock; motor runs continuously and timing cycle is controlled by solenoid-energized pawl which engages ring gear on planetary gear system to drive indicating pointers; another pawl engages to stop indicating pointers; models to meet Military requirements for humidity, salt spray and fungus resistance available.

For more data circle MD-37, Page 297

MINIATURE RELAYS 38

. . . for critical applications

Hart Mfg. Co., 110 Bartholomew Ave., Hartford, Conn.

This 3.76-oz relay is undamaged by shock in excess of 50*g*.

Designation: Series R.

Size: $1\frac{1}{16}$ in. diam, 2 in. high; weighs 3.76 oz.

Service: For ambient temperatures from −65 to 200 C; withstand vibration from zero to 1500 cycles at 15*g* or better; contact ratings to 7.5 amp at 30 v dc or 115 v ac with resistive loads; interelectrode capacitance less than 5 mmfd; electrode to case capacitance less than $2\frac{1}{2}$ mmfd; coil resistances to 50,000 ohm available; operating life of more than 100,000 cycles; resists dust and moisture.

Design: Four-pole, double-throw relay; hermetically sealed in metal container; solder-lug terminals.

For more data circle MD-38, Page 297

MAGNETIC DRAINPLUGS 39

. . .may be inspected without draining oil

Technical Development Co., 1228 *Cherry St., Philadelphia 7, Pa.*

Integral valve closes to prevent oil drainage when magnetic portion of assembly is removed.

Designation and Size: A-7118 has 1-18 NS threads and is 1.94 in. long; A-7114 has 1-14 NF threads and is 1.94 in. long; A-720 has $1\frac{1}{16}$-12 NS threads and is 1.94 in. long; A-722 has $\frac{7}{8}$-18 NF threads and is 1.84 in. long.

Service: Removes harmful ferrous particles from lubricant in gearboxes or crankcases; magnetic portion may be removed and inspected to detect excessive wear rates without draining oil; prevents loss of oil when gearboxes or crankcases are refilled, due to failure to replace plug.

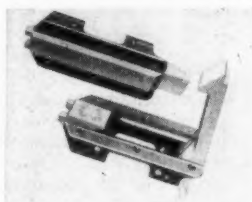
Design: Assembly incorporates valve which is automatically closed when magnetic plug is removed; draining attachment is inserted in place of plug to open valve for draining; attachment designated D-730 fits all four sizes of plug; plugs made from heat-treated aluminum alloy; synthetic rubber O-ring valve seals.

For more data circle MD-39, Page 297

FLUSH LATCH 40

. . . entirely of corrosion-resistant material

Hartwell Co., 9035 *Venice Blvd., Los Angeles* 34, *Calif.*



Latch mechanism fabricated from aluminum alloy extrusions has very high strength.

Designation: H-5200.

Size: 4.400 in. long, 2.000 in. wide, extends 0.855-in. back of door when latched; flush with 0.091 ±0.005-in. thick door; cutout for installation is approximately 3.390 by 0.780-in.

Service: Resists corrosion; easily unlatched; requires only simple rectangular cutout for mounting; has high strength.

Design: Pressure on trigger releases bolt which rotates on pin; bolt then extends approximately 2 in. above door surface and is used as handle to open door; trigger and bolt of 24S-T6 extruded aluminum alloy; bracket is type 302 stainless steel; springs are cadmium plated music wire; rivets are type 302 stainless steel or Monel; also available for other door sheet thicknesses.

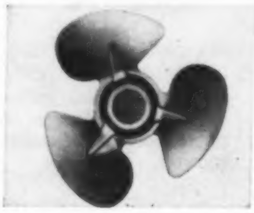
For more data circle MD-40, Page 297

NEW PARTS AND MATERIALS

FAN BLADES 41

. . . one piece, for unit bearing motors

Torrington Mfg. Co., Torrington, Conn.



Hand-set to assure accurate forming, blade alignment and smooth, quiet operation.

Designation: Series LU.

Size:

Fan No.	Fan Diam (in.)	Fan Depth (in.)	Average Weight (lb)	Fan No.	Fan Diam (in.)	Fan Depth (in.)	Average Weight (lb)
LU-7713-3	7¾	1¾	0.175	LU-8727-3	8¾	2¾	0.225
LU-7720-3	7¾	2	0.175	LU-8730-3	8¾	3	0.225
LU-7722-3	7¾	2¼	0.175	LU-1020-3	10	2	0.300
LU-7725-3	7¾	2½	0.175	LU-1022-3	10	2¼	0.300
LU-7727-3	7¾	2¾	0.175	LU-1025-3	10	2½	0.300
LU-8720-3	8¾	2	0.225	LU-1027-3	10	2¾	0.300
LU-8722-3	8¾	2¼	0.225	LU-1030-3	10	3	0.300
LU-8725-3	8¾	2½	0.225				

Service: LU-7713-3 delivers 390 cfm at 1550 rpm and requires 0.004-hp in NEMA free air test, delivers 145 cfm at 1550 rpm and 0.100-in. of water and requires 0.004-hp in NAFM code test; LU-1030-3 delivers 880 cfm at 1550 rpm and requires 0.026-hp in NEMA free air test, delivers 580 cfm at 1550 rpm at pressure of 0.100-in. of water and requires 0.021-hp in NAFM code test; ratings of other models fall between these values.

Design: Three-blade fans of mill-finish aluminum; carefully aligned and profiled to uniform contour but not statically balanced; pierced for all unit bearing motors; available in other pitches; available of steel on special order.

For more data circle MD-41, Page 297

SOLENOID CONTACTORS 43

. . . for ambient temperatures to 120 C

Guardian Electric Mfg. Co., 1621 *W. Walnut St., Chicago* 12, *Ill.*

Constructed in accordance with MIL-R-6106 specification, these contactors are hermetically sealed and extremely rugged.

Designation and Size: G-55335-A is 2$\frac{31}{32}$ in. by 3 in. by 3.406 in. high and weighs 1.3 lb; G-55759, G-54978 and G-54978-A are 2⅛ in. by 2$\frac{13}{16}$ in. by 4⅛ in. high and weigh approximately 2 lb.

Service: Maximum operating voltage, 29 v dc; pickup voltage, 18 v; dropout voltage, 7.0±5.5 v; coil current, 0.5-amp at 24-28 v dc; withstand 50*g* shock; not affected by dust or moisture; meet MIL, AN and JAN resonance requirements; for continuous operation;

Model	Resistive Load (amp)	Inductive Load (amp)	Motor Load (amp)
G-55335-A	50*	50*	50*
	25†	25†	25†
G-55759	100	50	100
G-54978	250	100	250
G-54978-A	200	100	200

* Normally open contacts. † Normally closed contacts.

Design: Single-pole, double-throw and single-pole, single-throw contactors; steel enclosures; black lusterless finish per Federal specification TT-C-595 units are normally grounded but can be furnished ungrounded; stud connections for contact and coil terminals; all contactors have mounting lugs.

For more data circle MD-43, Page 297

HEAVY-DUTY TERMINALS 42

. . . may be used with various wire types

Aircraft-Marine Products Inc., 2100 *Paxton St., Harrisburg, Pa.*

Characteristics of the crimp permit these terminals to be used with solid, stranded or irregularly shaped wires singly or in combination.

Size: For No. 8 through No. 4/0 wire.

Service: Terminating wires for connection with studs or screws; current-carrying capacity is high; grip on wire exceeds Underwriters' specifications; resist corrosion.

Design: Flag-type terminal made of pure copper and electrotinned; inner surface of barrel is indented to assure positive grip on wire; barrel seams are brazed to present even crimping surface; hand and power tools available for installing.

For more data circle MD-42, Page 297

POTENTIOMETER ELEMENT 44

. . . made of conductive plastic

Markite Corp., 155 *Waverly Place, New York* 14, *N. Y.*

Conductive plastic material has very high wear resistance, assuring long life with high accuracy.

Designation: Type 2028.

Size: 11 in. total length, 10 in. conductive length; ⅝-in. wide, 0.020-in. high base; 0.024-in. total height.

Service: For varying electrical resistance linearly; total resistance 20,000 ohm ±5%; linearity ±1%; resists vibration, wear and corrosion; tests run for 8000 to 10,000 operating cycles show retention of original resistance and linearity.

Design: Resistive track of conductive plastic molded in rigid asbestos-filled phenolic insulating base; base has same thermal expansion rate as resistive track; elements of higher precision available as specials.

For more data circle MD-44, Page 297

Industry's Widest Range

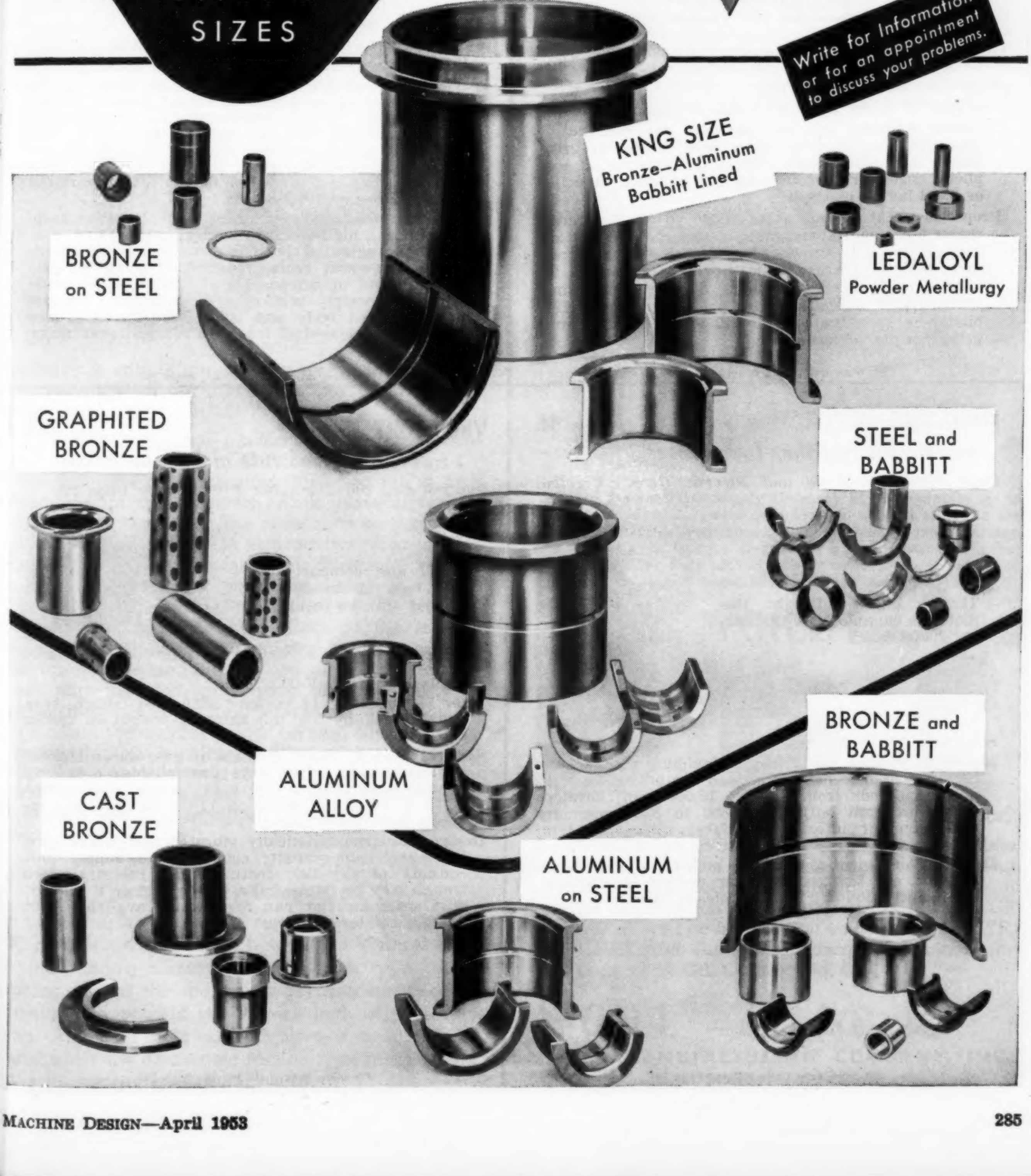
•• OF SLEEVE BEARING TYPES & SIZES

In this photo you will find a Johnson Sleeve Bearing of every type used in industry. Various alloys . . . bronze, babbitt or aluminum . . . can be selected . . . and different combinations with bronze or steel are available. Johnson Bronze facilities make possible a size range from tiny bearings up to King Size, 14" OD, 17 inches in length. Since Johnson Bronze produces all the types of sleeve bearings, their engineers can give you unbiased advice on the bearing best suited to your application.

JOHNSON BRONZE COMPANY
525 South Mill St. • New Castle, Pennsylvania

JOHNSON BEARINGS
Sleeve-Type

Write for Information or for an appointment to discuss your problems.



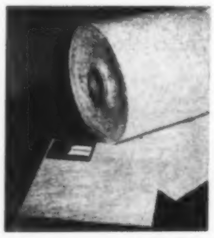
NEW PARTS AND MATERIALS

GLASS FIBER MAT 45

. . . for plastic reinforcement

Ferro Corp., Fiber Glass Div., 200 *Woodycrest Ave., Nashville* 20, *Tenn.*

Almost pure white, this mat may be used to reinforce plastics of very light shades without stains or discoloration where the mat is close to a product's surface.



Designation and Size: Uniformat GP and HSB; 50 in. widths;

Size Designation	Weight per Square foot (oz)	Roll Length (ft)	Area Per Pound (sq. ft.)
1½	1½	150	10.7
2	2	125	8.0
3	3	100	5.3

Form: Bonded glass fiber mat.

Service: GP is for matched metal-die plastic molding; resists washing and tearing; also for periodic or continuous dip impregnation; will retain integrity during and after impregnation; HSB is for atmospheric cure work produced by bag molding, tailoring or hand lay-up methods.

Properties: Mat is made from 14,000 yd per lb strands cut to 2 in. lengths and matted, strands each contain 204 continuously drawn filaments; mats have uniform weight, texture and color; GP binder is insoluble in styrene component of polyester resins, HSB binder is soluble; rate of solubility of HSB binder is such that mat retains its strength until gelling of plastic has started.

For more data circle MD-45, Page 297

AIR VOLUME BOOSTER 47

. . . has high volume output with low input

Kendall Controls Corp., 144 *Moody St., Waltham* 54, *Mass.*

Converts signal pressure to high volume output at pressure equal to signal pressure within one per cent.

Designation: Model 20.

Size: Has ¼-in. pipe threaded inlet, outlet and signal ports.

Service: Delivers air at pressure equal to or ½, ⅓, 2 or 3 times signal pressure; output pressure maintained within 1% of correct ratio to air pressure; will pass 35 cfm free air with 90 psi supply; will respond to signal pressure changes of ¼-in. of water or less; reverses from forward to relief flow with less than ±2 in. water detent; easily serviced without disconnecting piping.

Design: Diaphragm-pressure controlled air valve; deep-convolution, high-strength compensating and control diaphragms; 6 large exhaust ports; neoprene O-ring valve-seat seals; removal of 3 screws permits removal of screens, supply valve, inner relief valve, and supply seat ring for servicing; die-cast black-anodized body and bonnet; heat-treated and polished stainless-steel inner valves and seat rings.

For more data circle MD-47, Page 297

MINIATURE MOTOR 46

. . . weighs less than four ounces

Connecticut Telephone and Electric Corp., Meriden, Conn.

Though small and light, this motor is capable of withstanding 2000*g* shock.

Size: 1 by $1\frac{1}{16}$ by 2½ in.

Service: Operates on 12 v dc; maximum no-load current is 220 ma; maximum load current is 510 ma; no load speeds from 8000 to 12,000 rpm; develops 2 w maximum output at 6850 to 8350 rpm; for continuous or intermittent duty; withstands temperatures of −50 to 140 F.

Design: Permanent-magnet dc motor; die-cast housing and end bell; commutator molded to stainless-steel shaft; mounted by four holes in base.

For more data circle MD-46, Page 297

VIBRATORY FEEDER 48

. . . controls feed rate of bulk materials

Syntron Co., 260 *Lexington Ave., Homer City, Pa.*

Small and compact but will feed up to 500 lb of material per hour.

Designation: Model F-00.

Size: 7¼ in. high, 14 in. long with feed trough, base plate is 4⅛ by 5⅝ in.; shipping weight of feeder and controller is 32 lb.

Service: Feeding sand, powder or granular materials or small parts; feed rate controllable; operating voltage either 110 or 220 v ac; power consumption is 15 w when operating on 110 v 60 cycle ac; requires little maintenance.

Design: Electromagnetically vibrated trough and control-power supply unit; control-power supply unit consists of dry-disk rectifier and rheostat; feed trough may be flat, tubular, half round or V-shaped; dust seals for flat pan feeders are available; also available in larger sizes.

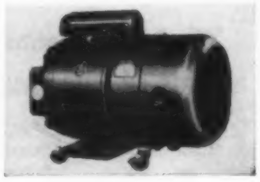
For more data circle MD-48, Page 297

NEW PARTS AND MATERIALS

AIR PUMPS 49

. . . keep system oil free

Romec Div., Lear Inc., Elyria, O.



Self - lubricating carbon-graphite composition pump rotor blades and thrust plates prevent oil from contaminating air.

Designation: Series RG.

Size:

Model No.	Length (in.)	Width (in.)	Height (in.)	Port Size (NPT, in.)	Weight (lb)
RG-5860	7	4	5½	¼	13.25
RG-5960	7	4	5½	¼	15
RG-5910	9$\frac{11}{16}$	6$\frac{1}{16}$	6$\frac{11}{16}$	¼	27.5
RG-5920	11⅝	6⅛	8⅜	¼	36.2
RG-5930	11$\frac{13}{16}$	6½	8⅛	⅜	39
RG-5940	11¾	6½	8⅛	⅜	41

Service: Pumps require no lubrication; no signs of failure after 5000-hr test runs; all pump parts easily accessible; may be used as vacuum or pressure pumps; discharge pressure is 10 psi for all; vacuum is 20 in. Hg for all; all motors are rated 115-230 v 60 cycle ac, 1725 rpm; pump wear taken up automatically;

Model No.	Capacity* (cfm)	Capacity† (cfm)	Motor (hp)	Model No.	Capacity* (cfm)	Capacity† (cfm)	Motor (hp)
RG-5860	0.5	0.1	1/20	RG-5920	2.6	0.6	1/3
RG-5960	1.0	0.2	1/12	RG-5930	4.0	..	1/3
RG-5910	1.3	0.3	1/4	RG-5940	5.0	..	1/3

*Free flow. †At 20 in. Hg vacuum.

Design: Pump-motor combination; motors are split-phase or capacitor start, single-phase; pumps assembled directly to motor shafts; pump rotor and four sliding blades are only moving parts.

For more data circle MD-49, Page 297

CIRCUIT SELECTOR SWITCHES 51

. . . for explosionproof service

G. H. Leland Inc., 123 *Webster St., Dayton* 2, *O.*

Container made of heavy-gage aluminum meets explosionproof requirements of MIL-E-5272, procedure 2.

Designation: Ledex.

Service: Selecting practically any number of electrical circuits; operate from 2 to 550 v dc; require power input of 60 to 100 w; wide variety of contact ratings available; circuit selection may be sequential, or instantaneous selection of desired circuit can be made; speed of operation is 35 to 40 steps per second when self-stepping; switch assembly may be located remotely from control position.

Design: Conventional ganged, wafer-type rotary selector switches driven by rotary solenoid; clips and rotors of switches are heavily silver-plated tempered brass; solenoid controlled by manually operated rotary switch or pushbuttons; all parts may be cadmium or chrome plated to meet military specifications; coil treated to prevent deterioration; wafer sections have 8, 10, 12 or 18 contacts; switches constructed to meet users specifications.

For more data circle MD-51, Page 297

TRANSMISSION BELT 50

. . . has increased flexibility

B. F. Goodrich Co., Akron, O.

Use of rayon fabric gives up to ten times greater flex life than that of belts using cotton fabric.

Designation: Drivesall.

Size: Any length to 500 ft; 3-ply belts are $\frac{7}{64}$-in. thick, weigh 5.7 lb per in. of width per 100 ft length and are available in 1, 1½, 2, 2½ and 3 in. widths; 4-ply belts are $\frac{11}{64}$-in. thick, weigh 7.9 lb. per in. of width per 100 ft length and are available in 2, 3, 4, 5, 6 and 8 in. widths; 5-ply belts are $\frac{13}{64}$-in. thick, weigh 10.0 lb per in. of width per 100 ft of length and are available in 6, 8 or 10 in. widths; 6-ply belts are $\frac{15}{64}$-in. thick, weigh 12.2 lb per in. of width per 100 ft of length and are available in 6, 8, 10 or 12 in. widths.

Service: Power transmission; resist moisture; stretch is 50% less than that of cotton belts;

Plies (No.)	Transmission Rating at 1000 rpm (hp)	at 2000 rpm (hp)	at 3000 rpm (hp)	at 4000 rpm (hp)	at 5000 rpm (hp)	at 6000 rpm (hp)
3.....	1.4	2.7	3.9	4.9	5.8	6.2
4.....	1.8	3.6	5.2	6.6	7.7	8.3
5.....	2.3	4.5	6.5	8.2	9.6	10.4
6.....	2.8	5.4	7.8	9.9	11.5	12.5

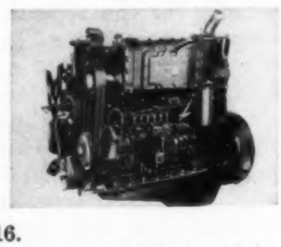
Design: Square-edged rayon fabric and rubber belts; plies are seamless; fabric is full width; tan in color with gray edges.

For more data circle MD-50, Page 297

DIESEL ENGINE 52

. . . compact and light, develops 165 hp

Buda Co., Harvey, Ill.



Overall length of this 6 cylinder engine is claimed to be within ½-in. of an aircooled V-8 diesel engine.

Designation: Model 6-DAS-516.

Size: 47½ in. long, 32$\frac{11}{16}$ in. wide and 47$\frac{7}{16}$ in. high; weighs 1825 lb with electrical accessories and air compressor; SAE No. 2 flywheel housing; 4$\frac{7}{16}$ in. bore, 5$\frac{9}{16}$ stroke, 516 cu in. total displacement.

Service: Maximum bhp is 165 at 2400 rpm; maximum torque is 402 lb-ft at 1600 rpm; rotation, clockwise, viewed from timing gear end.

Design: Overhead valve, 4-cycle, 6-cylinder, full diesel engine; crankshaft is fully counterbalanced heat treated forging, all journals Tocco hardened; removable wet sleeve cylinder liners; one-piece cast iron crankcase and cylinder block; steel backed copper-lead connecting rod and main bearings; aluminum alloy pistons with 5 rings; chrome-nickel steel inlet valves; exhaust valves have silichrome XB stems and austenitic steel heads; full pressure oiling system.

For more data circle MD-52, Page 297

ENGINEERING DEPARTMENT

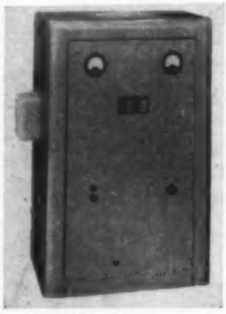
EQUIPMENT

For additional information on this new equipment, see Page 297

PRECISION POWER SUPPLY 53

. . . uses three-phase, high-voltage input

Inet Inc., 8655 S. Main St., Los Angeles 3, Calif.



High power dc source for digital and analog computers, business machines and precision electronic equipment.

Designation: Type RR-200-15, Style FSPPSF.

Service: Providing closely regulated high voltage, high current dc power; regulation, 0.2%; ripple, 0.01% rms if desired —0.5% standard; low maintenance because of no moving parts and no vacuum tubes; compact size; output, 15 amp at 200 v dc; input, 208 v ac ±10%, 3-phase, 60 cycles ±3 cycles.

Design: Selenium rectifier, magnetic amplifier regulated; equipment available for 230 and 460 v 3-phase input power; 14 gage steel cabinet optional; units available in indoor floor-standing cabinets, caster-mounted cabinets or for relay rack mounting.

For more data circle MD-53, Page 297

SERVO TESTER 54

. . . provides quick analysis of servo loops

Industrial Control Co., Wyandanch, L. I., N. Y.

Servo response shown on CRO screen is compared to manufacturer's specifications drawn on transparent mask.

Designation: 101-A.

Service: Checking, adjusting and maintaining servo systems using no additional instrumentation; special masks are prepared with outline of correct transient response and numbers describing test conditions, e.g., peak output displacement, sweep period, carrier phase and frequency; front panel controls are set by mask data; tester connected by a standard cable to a servo test receptacle on equipment; a forcing voltage, injected into error channel through this cable, drives loop into a periodic transient; output transient appears on 3 in. cathode ray oscillograph screen; correspondence between screen trace and mask outline is used for acceptance, adjustment or replacement of servo; period—0.2, 1 and 5 seconds; y-axis sensitivity, 0.5 v dc per in. max; power requirement—117 v, 60 or 400 cps, 45 w.

Design: Transient signal generator circuit with synchronized sweep circuit.

For more data circle MD-54, Page 297

HAND POLARISCOPE 55

. . . detects glass or plastic strains

Pacific Transducer Corp., 11921 *W. Pico Blvd., Los Angeles 64, Calif.*

Objects are examined between polarizer and analyzer using any moderate-intensity light source.

Designation: 242.3.

Size: 3 in. diam; 3¼ in. space between filters.

Service: Detecting strains in glass, clear plastics and glass-to-metal seals; has fixed polarizer at one end of frame; analyzer, rotatable through 180 deg, is located at other end of frame.

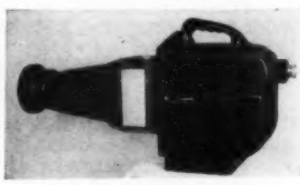
Design: Polaroid filters; nonrusting oxidized metal frame; hand-held.

For more data circle MD-55, Page 297

SEQUENCE CAMERA 56

. . . produces large pictures at high speed

Charles A. Hulcher Co. Inc., 40 *Manteo Ave., Hampton, Va.*



Takes pictures on 7-mm film at rates up to 25 five-inch frames per second or 50 two and one-half inch frames per second.

Designation: Hulcher 70.

Size: Without lens, 16.67 in. long, 10.75 in. wide, 13.00 in. high; weight, 32 lb.

Service: Producing pictures as large as stills in sequence as fast as movies of parts or units in motion; film is held in flat plane and is not in motion during period of exposure; also takes stereo pictures by inserting two lenses in lens plate; stereo pictures taken at speeds up to 25 double frames per second with each picture 2¼ x 2½ in.; 1/1200-second film exposure; focusing provided by means of reflex type viewfinder; power requirement—110 v ac or dc.

Design: Double rotating disk-type shutter; no lens is supplied but is equipped with lens plate to fit 7-in. focal length Eastman Kodak Aero Ektar f/2.5 lens; can be adapted for wide range of lenses; 28 v dc models available.

For more data circle MD-56, Page 297

Facts about *HELI-COIL* inserts you should know

What they are

*Heli-Coil** screw thread inserts are precision formed coils of stainless steel or phosphor bronze wire. Wound into tapped holes, they form permanent, non-corrosive, strip-proof threads of astonishing strength. Available for National Coarse, National Fine and Unified threads, pipe threads and spark plug threads. They are made in all standard sizes and lengths for assemblies requiring Class 3, 3B, 2 or 2B fits.

What they are for

AS ORIGINAL COMPONENTS: *Heli-Coil* inserts are used to provide stronger, lighter fastenings, corrosion-proof, wear-proof threads in all assemblies.

FOR PRODUCTION SALVAGE: When conventional tapped holes are damaged in production, restore them on the line with *Heli-Coil* inserts. Get better-than-original strength with no increase in screw size and no tell-tale signs of rework.

FOR SPEEDY REPAIRS: When tapped threads wear, strip or corrode in service, renew them in minutes on location in shop or field with *Heli-Coil* inserts. No welding—no plugging—no secondary machining—no oversize screws.

How they work

Holes are drilled and tapped as you do for ordinary threads—then *Heli-Coil* inserts are wound into tapped holes by hand or power tools. Install in a few seconds, assure thread protection forever. Can be used in any metal wood or plastic.

No other method is so simple, effective and practical.

What they do for you

Heli-Coil inserts *save money* because they strengthen threads and make *fewer* smaller fastenings do the same holding job. They make *lighter* bosses and flanges practical and they *save weight* in two ways: (1) by permitting use of cap screws, instead of bolts and nuts; (2) by allowing use of smaller, shorter, fewer cap screws. *Heli-Coil* inserts *protect* your product from thread wear, galling and stripping for life in every kind of metal, in plastics or wood. They preserve customer good-will by preventing product failure, due to thread fault. *Heli-Coil* inserts improve the end product, cut rejects, salvage threading errors.

Best time to put *Heli-Coil* inserts benefits to your use is right at the designing board, as many leading manufacturers are doing. But to convince you of their many advantages ask for a working demonstration right on your production line. Write today! Complete information and engineering data is available in the *Heli-Coil* catalog. Use Coupon!

*Reg. U.S. Pat. Off.

Approved for All Military and Industrial Uses

7 savings wrapped up in HELI-COIL inserts

These savings are multiplied into major profits this simple way

Use *Heli-Coil** Inserts to:

1. *Save* money by using fewer and smaller screws to do the same holding job.
2. *Save* material—lighter bosses, thinner wall section, smaller flanges.
3. *Save* weight and reduce bulk in assemblies.
4. *Save* assembly time by using cap screws instead of nut-and-bolt assemblies.
5. *Save* rejections in production. Threads damaged on the line are quickly repaired. You save time. Reduce scrap.
6. *Save* on field service costs. No field damage to threads—fortified by *Heli-Coil* inserts.
7. *Save* customer good will by eliminating product failure due to thread fault. Every thread in your product is made stronger, longer wearing with *Heli-Coil* insert protection.

Use the handy coupon to get *free sample Heli-Coil* inserts plus all the data you need to design these savings into your product.

HELI-COIL CORPORATION
124 SHELTER ROCK LANE, DANBURY, CONN.

☐ **Send samples and Handbook 652, a complete design manual.**
☐ **Send samples and put my name on list to receive "Heli-Call," case history periodical.**

NAME__________ TITLE__________

COMPANY__________

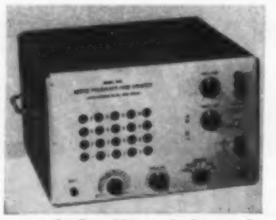
ADDRESS__________

CITY__________ ZONE_____ STATE__________ 2012

FREQUENCY-TIME COUNTER 57

. . . has speed, accuracy and simplicity

Potter Instrument Co., 115 Cutter Mill Rd., Great Neck, N. Y.



Small low-cost instrument with wide range of measurement facilities.

Designation: 840.

Size: 17⅛ in. wide, 10½ in. high, 16¾ in. deep; weight, 55 lb.

Service: Measuring frequencies, periods, time intervals and frequency ratios; use as totalizing counter and direct reading rpm tachometer; direct frequency measurement, 0 to 100 kc; period measurement, time for 1 or 10 cycles; time interval measurement and relay contact timing range, 10 microseconds to 100,000 seconds; measures ratios of two external frequencies; secondary frequency standard for 1 cps, 10 cps, 100 cps, 1 kc and 100 kc; counter rates, up to 150 kc; input sensitivity, 0.2-v min; power requirement—105-125 v, 50-60 cycles, 250 w.

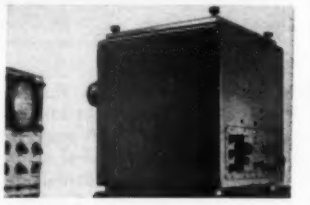
Design: Five-decade electronic counter with input, gating and switching circuitry; 100 kc crystal oscillator; crystal-controlled time base; accessory connector for photoelectric or electromagnetic pickup; several models and modifications available.

For more data circle MD-57, Page 297

OSCILLOSCOPE RECORDING CAMERA 59

. . . has three modes of operation

J. A. Maurer Inc., Photographic Instrumentation Div., 37-01 31st St., Long Island City 1, N. Y.



Provides recording in the forms of still picture, continuously moving film, and drum photography.

Designation: M.731.

Size: 13 in. wide, 13 in. high, 12 in. deep.

Service: Providing records for measurement of phenomena displayed on a cathode ray tube; continuous feed mechanism provides for the movement of 25 ft of sensitized material at rates from 0.4 to 100 in. per second; drum attachment, carrying 20 in. of sensitized material, provides linear speeds from 4 to 1200 in. per second; perforated or unperforated film or paper of either 35 mm or 70 mm width may be used; 25 runs are possible for drum operation without removal of drum because of 50 ft film capacity magazine; shunt wound, 6-volt driving motor provides wide range of speeds.

Design: Electrical contacts provided to trigger oscilloscope in order that drum type record is made for only one revolution of drum; second electrical circuit permits synchronization of external events; standard lens is 3-in. focal length f/1.9; other lenses available.

For more data circle MD-59, Page 297

METER MATCHER 58

. . . reduces test-circuit loading

Keithley Instruments, 3868 Carnegie Ave., Cleveland 15, Ohio

Practically eliminates voltage and power measurement errors caused by the meter currents.

Designation: Model 105.

Size: 17 in. wide, 9 in. high, 10 in. deep; weight, 45 lb.

Service: Matching voltmeters or wattmeters to circuits under test to reduce loading; aids in measurement of synchro outputs and other linear and nonlinear circuits affected by meter currents; test circuit is connected to input of meter matcher and output is connected to a voltmeter or voltage coil of a wattmeter; input current, 150 microamp max; ac input voltages—15, 75, 150, 300, and 600; output current, 0.05 to 0.07-amp; max ac output voltage, 150; output impedance, equivalent to 1 ohm in series with 15,000 microfarads; frequency response, 5 to 20,000 cps; power requirement—110-120 v, 50-60 cps, 200 w.

Design: Power frequency amplifier with voltage divider input.

For more data circle MD-58, Page 297

ELECTRONIC RECORDER 60

. . . used with analog computers

Goodyear Aircraft Corp., Akron 16, Ohio

Provides for remote operation of computer systems by built-in control units.

Designation: R5 GEDA.

Size: Desk height.

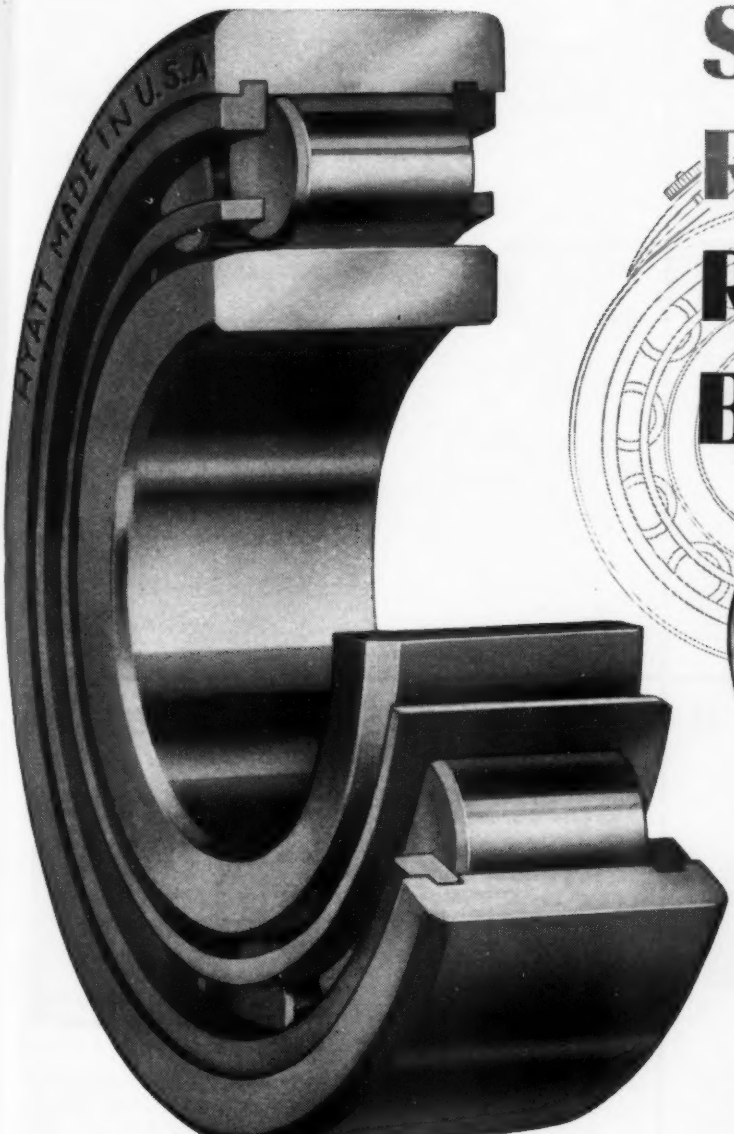
Service: Recording data from analog computers; six recording channels are provided either in curvilinear co-ordinates or rectilinear co-ordinates as operator desires; operator may sit at console which has ample writing and storage space for records and logs; sensitivity, continuously variable from 0.01 to 100 v per mm; input impedance, 2.5 megohms on 0.01 to 0.1-v per mm range and 10 megohms on all other ranges; frequency response, essentially flat to 100 cps when ink pens are used; accuracy limited only by linearities of pen motors; power requirement—105 to 125 v, 60 cycles, 350 w.

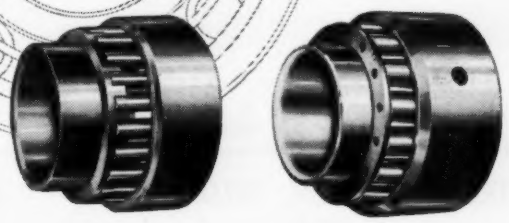
Design: Electronic recorder using six dc amplifiers; ink pens used on rectilinear co-ordinates; desk-type console.

For more data circle MD-60, Page 297

Answer to a LOADed question . . . the best way to handle radial loads is with . . .

HYATT STRAIGHT RADIAL ROLLER BEARINGS





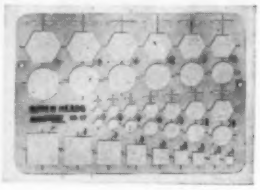
Wherever the jobs . . . and the loads . . . are big, you'll find radial roller bearings smoothing the path of power, because radial roller bearings give the greatest load-carrying capacity within given boundary dimensions. And, whenever radial loads enter the picture, design engineers naturally turn to Hyatt for the most complete line of radial roller bearings available anywhere. The wide range of Hyatt Roller Bearing sizes and types allows the engineer greater design flexibility, makes his job easier. If you are not familiar with the complete Hyatt line, write for our general catalog No. 150. Hyatt Bearings Division, General Motors Corporation, Harrison, N. J.

HYATT ROLLER BEARINGS

PLASTIC TEMPLATE 61

. . . for hex or square screw heads and nuts

Rapidesign Inc., P.O. Box 592, Glendale, Calif.



Speeds the drawing of top views as well as side views of screw heads and nuts.

Designation: 57.

Size: $7\frac{3}{8}$ in. wide, 5 in. high, 0.030-in. thick.

Service: For general purpose drafting; covers 13 hexagon screw heads and nuts from $\frac{1}{4}$-in. to 1 in. across flats in $\frac{1}{16}$-in. increments; provides 9 squares for square screw heads and nuts in $\frac{1}{16}$-in. increments from $\frac{1}{4}$-in. to $\frac{1}{2}$-in. across flats and in $\frac{1}{8}$-in. increments from $\frac{1}{2}$-in. to 1 in. across flats; pencil allowance for accuracy.

Design: Cutouts arranged so that hexes, chamfer circles, flat and screw body diameters can be drawn for each size and elevation; cutouts are precision smooth; made of matte finish plastic.

For more data circle MD-61, Page 297

OSCILLOGRAPH TABLE 63

. . . movable and has tiltable top

Allen B. Du Mont Laboratories Inc., 1500 *Main Ave., Clifton, N. J.*

Ease of viewing cathode-ray oscillographs from either sitting or standing position is provided.

Designation: Type 2602.

Size: 19 in. wide, $36\frac{5}{8}$ in. high, 31 in. deep; weight, 57 lb.

Service: Movable table for mounting cathode-ray oscillographs in tilted position; top tilts from horizontal plane to 20 deg angle; oscillographs can be placed at varying depths on tilted top with adjustable bar which supports table; contains shelf and drawer for tools, auxiliary instruments and components.

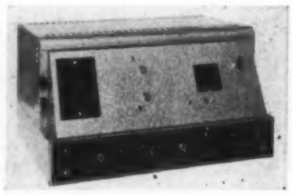
Design: Cold-rolled steel construction; standard Du Mont gray wrinkle finish; chrome supporting members; rubber-tired swivel casters.

For more data circle MD-63, Page 297

ELECTRO-MECHANICAL INTEGRATOR 62

. . . accurate, high speed and stable

Instron Engineering Corp., 2 *Hancock St., Quincy* 71, *Mass.*



Registers the time integral of any dc input function on a mechanical counter.

Designation: I-101, I-102.

Size: 17 in. wide, 12 in. high, 10 in. deep; weight 35 lb.

Service: Integrating stress-strain curves, analyzing graphical data, integrating rocket thrust, continuous process weighing, totalizing moisture determinations in paper, textile and chemical products; uses standard computor principles; input controls a variable speed motor whose total shaft rotation becomes a direct measure of the time integral of input function; integrated value is registered on mechanical counter; integrating rate, 5000 counts per minute; frequency response, 0 to 5000 cps; full scale input signal, ± 50 v dc; input impedance, essentially infinite; linearity, 1%; power required—110 v, 60 cycles, 50 w.

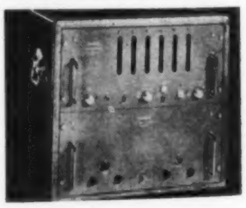
Design: Velocity servo circuit controlling a variable speed motor coupled to a dc tachometer; electronic integrator; stroboscopic calibration timer; resettable mechanical revolution counters; single or double counter models available; 50 cycle power requirement models available.

For more data circle MD-62, Page 297

EVENTS-TIME METER 64

. . . counts to 1,000,000 pulses per second

Berkeley Scientific Co. Div., Beckman Instruments Inc., 2200 *Wright Ave., Richmond, Calif.*



High-speed electronic counter records the number of events occurring during a precise time interval.

Designation: Model 5558.

Size: $20\frac{3}{4}$ in. wide, 19 in. high, 15 in. deep; weight 120 lb.

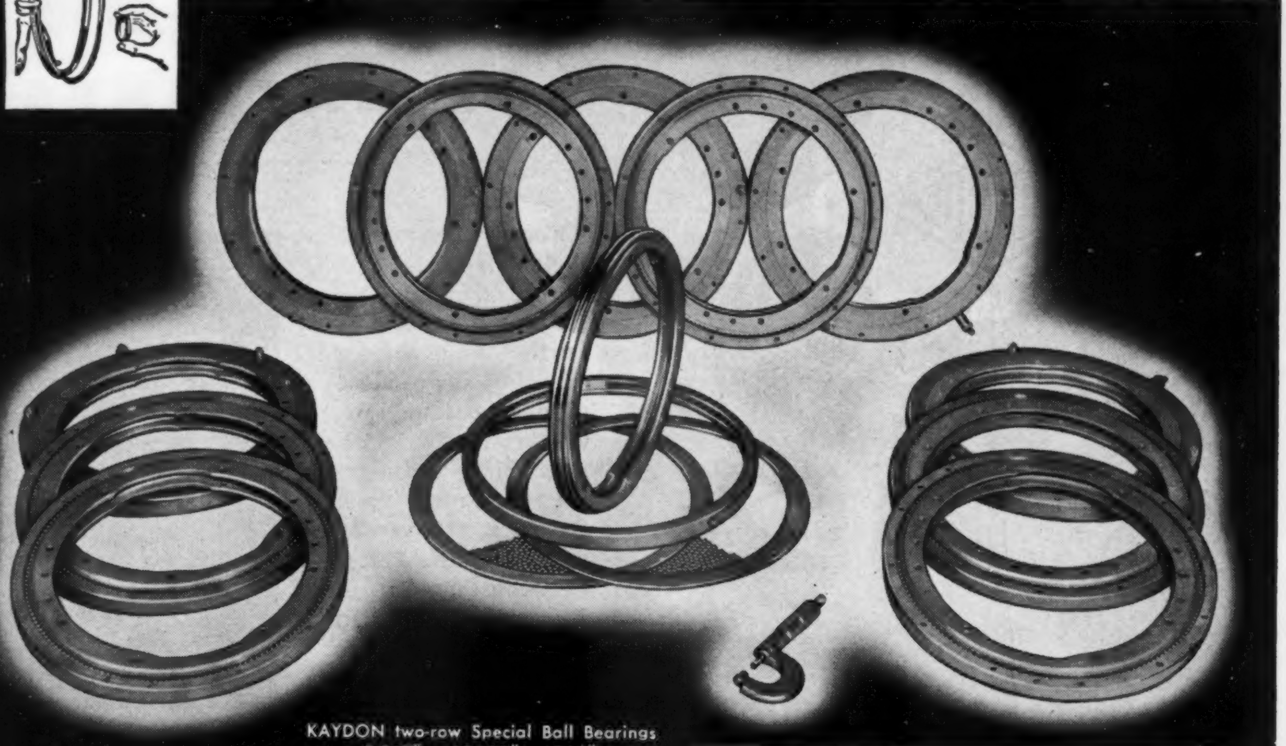
Service: Counting events occurring either regularly or with random distribution at rates from 20 to 1,000,000 times per second; result displayed in digital form on illuminated number panels of electronic counting units; operates automatically or manually; when operated automatically, result is displayed from 1 to 5 seconds then recycles; when operated manually, result is displayed until reset button is depressed to obtain another reading; in "test" position complete check of unit can be made in 2 seconds; also operates as straightforward counter; accuracy, ± 1 event; stability, 1 part in 10^6; power requirement—105-130 v, 50-60 cycles.

Design: Input circuit, electronic gate which is opened and closed by crystal controlled time base; electronic counters and output circuitry; several modified units available.

For more data circle MD-64, Page 297

Contact KAYDON of Muskegon

FOR ALL TYPES OF BALL AND ROLLER BEARINGS: 4" BORE TO 120" OUTSIDE DIAMETER



KAYDON two-row Special Ball Bearings
8.062" x 10.687" x .750"

Dependable Life-Savers ...KAYDON-bearinged Piasecki HELICOPTERS

CONTROL movements of Piasecki HUP Helicopters . . . life-savers of the air . . . are transmitted to the spinning rotor blade assemblies by means of a swash plate that moves on unique two-row KAYDON Special Ball Bearings.

Control of the helicopter is obtained by the movement of its rotor blades. Each blade can move in six different directions while they are rotating. Thus the mechanism responsible for control movements is complex, important, and it demands the utmost in bearing-precision.

KAYDON met the challenge of this intricate bearing-problem with these special two-row, thin section, 8.062" x 10.687" x .750" ball bearings. Similarly, KAYDON cooperates with designers of many types of precision equipment to achieve their objectives.

We specialize in large, thin-section, light weight precision bearings for Aircraft, Automotive and Industrial Equipment

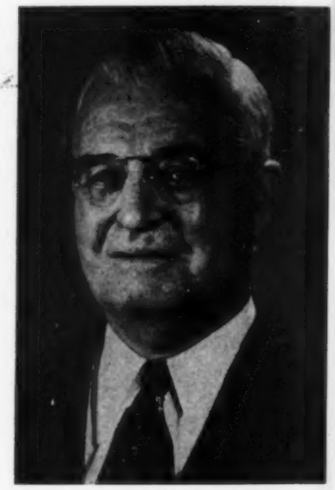
KAYDON Types of Standard and Special Bearings:
Spherical Roller • Taper Roller • Ball Radial • Ball Thrust • Roller Radial • Roller Thrust • Bi-Angular Bearings

THE KAYDON ENGINEERING CORP.

MUSKEGON • MICHIGAN

PRECISION BALL AND ROLLER BEARINGS

MEN
OF MACHINES



Myron S. Curtis

Former director of engineering at Warner & Swasey Co., Cleveland, **Myron S. Curtis** has been named vice president of engineering. Mr. Curtis received his mechanical engineering degree from Brown University and joined the Potter & Johnston Co. of Pawtucket, R. I. In 1939 he resigned to become head of the Shell Lathe Development Project which was undertaken by the National Machine Tool Builders' Association and the Army Ordnance Department. He joined the engineering department of Warner & Swasey the following year. A member of the planning committee formed to guide the company's new product development, he has been largely responsible for the development of the Sulzer weaving machine. Named assistant director of engineering in 1945, he became director of engineering and a member of the board of directors three years later. Mr. Curtis has been awarded many patents in the machine tool field, and during World War I he received a citation from the War Department for the engineering and management of a plant for production of shells.

♦

Angus A. MacDonald has been appointed assistant chief engineer in charge of two-way radio development in the Communications and Electronic Div. of Motorola Inc., Chicago. He will head a group in the development of mobile two-way radio equipment for use in public safety, land transportation, industrial and related fields. Before joining Motorola, Mr. MacDonald was a section manager with the Westinghouse Electric Corp.

♦

New staff engineer in charge of engineering on the company's program on aircraft hydraulic equipment, **Kilbourne Knox Jr.** recently joined Parker Appliance Co., Cleveland. He was a project engineer and preliminary designer with Spartan Aircraft Corp. and with Curtiss-Wright Corp., served as vice president

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and chief engineer of the Globe Aircraft Corp. and, until his new appointment, was senior designer on hydraulic equipment with the McDonnell Aircraft Corp.

•

Werner I. Senger

The board of directors of Gisholt Machine Co., Madison, Wis., has elected **Werner I. Senger** to the newly created position of vice president in charge of balancing. Mr. Senger has been associated with the company for over 35 years and has been in charge of balancing machine development and engineering since 1924. He received his bachelor of science degree in mechanical engineering from the University of Wisconsin and his master's degree from Yale University. Recognized as a leading authority on balancing, Mr. Senger has written a number of treatises on the subject, including several articles which have appeared in MACHINE DESIGN.

•

Curtis L. Bates has joined Ryan Aeronautical Co., San Diego, as assistant director of engineering. For ten years he has been associated with Northrop Aircraft Inc., and prior to that time served at Douglas Aircraft and as chief engineer of the Aero Industries Technical Institute. He also spent some time in Sweden as structures engineer on Swedish military airplane developments.

•

J. F. Weiffenbach has been appointed to the position of chief product engineer of the manufacturing division of Fairbanks, Morse & Co., with headquarters in Chicago. He was formerly vice president in charge of manufacturing of the Canadian Locomotive Co. Ltd.

•

Included among appointments made recently in the Motor and Generator Div. of General Electric are those of **Charles J. Koch** and **Bascom H. Caldwell**, named managers of engineering in the medium induction motor department and the synchronous and specialty motor and generator department, respectively. Mr. Caldwell will be located at the Lynn, Mass. works. He joined GE in 1935 after serving on the faculty of the University of Texas where he received his master's degree in electrical engineering. In 1946 he was named section engineer of dc motors and generators and became assistant manager of engineering four years later. He was manager of Lynn motor engineering at the time of his new appointment. Mr. Koch was among the first graduates of the co-operative course

in electrical engineering organized by MIT and GE. In 1924 he received his master's degree and joined the company on the test course in the induction motor engineering department. In 1939 he was appointed assistant designing engineer and in 1947 was named manager of the induction motor engineering section.

♦

Frank L. Egan has joined Fischer and Associates, Cleveland, as project engineer. He has served in various engineering capacities in government and industry and as a consultant.

♦

Formerly chief engineer, **G. G. Crewson** is now director of engineering of the Buffalo Electro-Chemical Co. Inc., a division of Food Machinery and Chemical Corp. **J. N. Vermilya** succeeds him as chief engineer, and **Charles M. Standart** has been appointed assistant to the chief engineer.

♦

William A. Reich has been appointed manager of advance development engineering at Carboloy Department of General Electric Co., Detroit.

♦

New chief engineer of the receiver division of Allen B. Du Mont Laboratories Inc. is **Alfred Y. Bentley,** who has been chief engineer of the cathode-ray tube division since 1947. Prior to that time he was assistant head of the cathode-ray tube engineering department, a position to which he was assigned upon joining the company in 1945. Mr. Bentley replaces **Robert J. Cavanagh,** who returns to his original engineering and research post with the company's Research Div.

♦

John P. Selberg has been appointed chief engineer of the newly formed Petro-Mechanics Research Div. of Borg-Warner Corp., to be located at North Hollywood, Calif.

♦

Lear Inc., Grand Rapids, Mich., has appointed **T. K. Greenlee** as chief electro-mechanical engineer. The Electro-Mechanical engineering operation of the Grand Rapids plant was recently established as a separate department to accommodate expansion in the field of aircraft actuating and control equipment. Mr. Greenlee was formerly chief engineer of Barber-Colman Co., in charge of actuator and controls development. He was responsible for the development of the company's aircraft engineering division.

♦

Elliot Schick has joined the engineering staff of Ebert Electronics Co., Hollis, N. Y. He formerly served as chief industrial engineer of the Emerson Radio & Phonograph Corp.

♦

Paul D. Cornelius has been appointed head of the design and research department of Vlier Engineering Inc., Los Angeles.

THE ENGINEER'S Library

Recent Books

Elasticity in Engineering. *By Ernest E. Sechler, professor of aeronautics, California Institute of Technology; 429 pages, 6 by 9 inches, clothbound; published by John Wiley & Sons Inc., New York; available from* MACHINE DESIGN, *$8.50 postpaid.*

An advanced mathematical treatment, this text deals with the fundamental theories of stress and deformation of elastic bodies under load. Choice of text material has been determined largely by the needs of aeronautical structural engineers; however, the presentation is intended to be of use to engineers in all phases of structural analysis. Fundamental equations and assumptions underlying the whole field of elasticity are established in the first part of the book. In the second part, the use of these principles in solving the elastic problems of stable (nonbuckling) structures is illustrated. The third and concluding part is devoted to the problems of unstable (buckling) elastic structures.

•

Electronic Engineering Principles. *By John D. Ryder, professor of electrical engineering, University of Illinois; 515 pages, 6 by 8½ inches, clothbound; published by Prentice-Hall Inc., New York; available from* MACHINE DESIGN, *$9.00 postpaid.*

This second edition introduces new and timely material into a fundamental and thorough treatment of electronics. An expansion of the feedback section and a new chapter on solid-state devices and the transistor have brought this text in line with modern trends and needs.

•

Advanced Mechanics of Materials. *By Fred B. Seely, professor emeritus, and James O. Smith, professor, theoretical and applied mechanics, University of Illinois; 698 pages, 6 by 9 inches, clothbound; published by John Wiley & Sons Inc., New York; available from* MACHINE DESIGN, *$8.50 postpaid.*

A second edition, this textbook is far more comprehensive than the first which appeared 20 years ago. Including developments of this period, the book is intended primarily for a second course in mechanics of materials. Topics and methods of presentation are adaptable to the needs of design and research engineers. The book provides a complete reference of methods used and results obtained in the

analysis of stresses in structural and machine members.

♦

Theory of Elasticity and Plasticity. *By H. M. Westergaard, professor of civil engineering, Harvard University; 190 pages, 5½ by 8½ inches, clothbound; published by John Wiley & Sons Inc., New York; available from* MACHINE DESIGN, $5.00 *postpaid.*

This book provides an introduction to the theories of elasticity and plasticity. Although the treatment is theoretical, it is written from the engineer's viewpoint. Fundamental concepts and mathematical relationships are presented with respect to stress, strain, displacement, Hooke's law, and the basic equation of elasticity. Other topics are Lame's stress ellipsoid, Mohr's circles, octahedral stress and strain, compatibility, cylindrical co-ordinates, and the laws of plasticity. Methods of application are offered and such tools as the strain potential, the Galerkin vector and the twinned gradient are demonstrated by examples. Subjects also discussed are hollow cylinders and spheres, rotating disks, thermal stresses, the problems of Kelvin, Boussinesq, Corruti and Mindlin as well as items involving cavities and deflections of surfaces.

♦

High Speed Photography. *By George A. Jones; 327 pages, 5½ by 8½ inches, clothbound; published by John Wiley & Sons Inc., New York; available from* MACHINE DESIGN, $6.50 *postpaid.*

Fundamentals, current practices and scope of high-speed photography are summarized. Scientific, commercial and industrial applications are pointed out.

♦

Color in Business, Science, and Industry. *By Deane B. Judd; 411 pages, 6 by 9 inches, clothbound; published by John Wiley & Sons Inc., New York; available from* MACHINE DESIGN, $6.50 *postpaid.*

Often obscured by abstract discussion, the subject of color is treated in practical terms in this book. It deals with the commercial and scientific aspects of color and its applications. Basic facts are presented in Part I concerning the eye, color terminology, color matching and color deficiencies. In Part II the tools and techniques for measuring and specifying color are discussed. Part III covers the physics and psychophysics of colorant layers.

Association Publications

Proceedings of the Sixteenth Annual National Time and Motion Study and Management Clinic. 168 *pages, 8½ by 11 inches, paperbound; available from Industrial*

Management Society, 35 *E. Wacker Drive, Chicago* 1, *Ill.;* $4.00.

Complete transcripts of talks by leaders of labor, management and government on various topics including time study, motion economy, methods, plant layout, materials handling, wage incentives, maintenance, and human relations are included in this booklet.

•

Technical Data on Plastics. 192 *pages,* 8½ *by* 11 *inches, clothbound; available from Manufacturing Chemists Association Inc.,* 246 *Woodward Bldg., Washington* 5, *D. C.;* $2.50.

Commercially available plastics are cataloged and described in this handbook. Characteristic properties of these materials and the variables which affect them permit use of the data for engineering design. Data are published for the first time on the alkyd and silicone molding compounds and epoxy resins. Two new sections show the properties of various plastics when made in the form of foams or thin films.

•

Compressed Air Power in Industrial Production. 36 *pages,* 8½ *by* 11 *inches, paper cover; available from Compressed Air and Gas Institute,* 1410 *Terminal Tower, Cleveland* 13, *Ohio;* 25 *cents.*

Third in a series, this pamphlet discusses pneumatic tools and uses of compressed air in production. It is directed to designers, engineers and production personnel.

•

Symposium on Determination of Elastic Constants. 104 *pages,* 6 *by* 9 *inches, paperbound; available from American Society for Testing Materials,* 1916 *Race Street, Philadelphia, Pa.;* $2.00.

This booklet contains five ASTM papers which discuss the problems involved in the determination of elastic constants and their extension to extreme temperature in metallics and nonmetallics. Measurement techniques for relatively new materials such as plastics and composite materials are included.

•

Welding Processes and Procedures Employed in Joining Stainless Steels. 52 *pages,* 8¼ *by* 11¼ *inches, paperbound; available from American Welding Society,* 29 *West 39th St., New York* 18, *N. Y.;* $2.00.

Information concerning metal-arc welding, carbon-arc welding, gas welding, resistance welding and flame cutting of stainless steels are compiled in this hand-

book-type review. The welding characteristics, advantages and limitations of various welding processes and recommended procedures are discussed in detail.

Manufacturers' Publications

Hydraulics as Applied to the Machine Tool Industry. *197 pages, 8½ by 11 inches, heavy paper-bound; available from Product Service Dept., Publications Section, Vickers Inc., 1430 Oakman Blvd., Detroit 32, Mich.;* $2.00.

One of a series prepared originally by the Henry Ford Trade Schools, this manual includes information about hydraulic pumps and valves and their maintenance and repair. Practical application of hydraulics to machines for the control of speeds and feeds is also covered.

♦

Kodak Industrial Handbook. *Multiple ring binder with four 5½ by 8½-inch paperbound sections—84, 64, 72, and 76 pages; available from Sales Service Div., Eastman Kodak Co., Rochester 4, N. Y.;* $4.00.

Four sections describing methods of photocopying, microfilming, photographing through a microscope, and making industrial pictures are included in this handbook. Provision is made for expansion to accommodate additional sections.

New Standard

Standards for Steam Surface Condensers. *28 pages, 8½ by 11 inches, paperbound; available from Heat Exchange Institute, 122 E. 42nd St., New York 17, N. Y.;* $2.00.

Definitions of essential terms and standards of heat transfer rates as well as performance expectations on condensers are covered. The information contained can be used for specifying or designing surface condensing equipment.

Government Publications

NACA Technical Series. *Each publication is 8 by 10½ inches, paperbound, side-stapled; copies available from National Advisory Committee for Aeronautics, 1924 F St., N.W., Washington 25, D. C.*

The following Technical Notes are available:

2812. **Effects of Cyclic Loading on Mechanical Behavior of 24S-T4 and 75S-T6 Aluminum Alloys and SAE 4130 Steel—53 pages.**

2820. **An Analysis of the Errors in Curve-Fitting Problems with an Application to the Calculation of Stability Parameters from Flight Data—29 pages.**

2821. **Torsion Tests of Aluminum-Alloy Stiffened Circular Cylinders—38 pages.**

2822. **A Special Investigation to Develop a General Method for Three-Dimensional Photoelastic Stress Analysis—59 pages.**

2840. **Buckling of Low Arches or Curved Beams of Small Curvature—75 pages.**

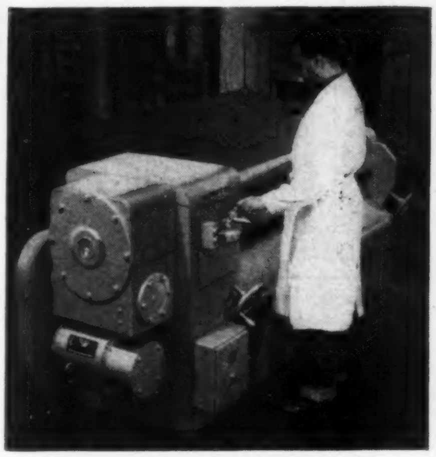
Advertisement

Brakemotor Prevents Drift, Saves Space on Niagara Bending Rolls

Use of a motor incorporating a spring-applied, electromagnetically released brake as an integral part of the motor design is an important factor in the simple, positive finger-tip control of the bending rolls produced by Niagara Machine & Tool Works, Buffalo, N. Y.

In the neutral position, rolls are instantly stopped by the action of the Brakemotor. This feature prevents drifting or overcoasting of the work in the rolls and thus permits more accurate operation. The fast, precise action of the brake allows work in the rolls to be reversed rapidly and frequently, without kicking out the thermal overload relays, as might occur in reversing by plugging.

The Brakemotor also permits jogging in short increments. The operator has complete and accurate control at all times.



Precise control of this Niagara Bending Roll is made possible by the Star-Kimble Brakemotor shown at lower left of machine

Saving in Frame Sizes

The Brakemotors used on these bending rolls are specially engineered to meet the speed and torque requirements of the application. As a result, Niagara has been able to use smaller frame sizes than would be possible with conventional designs.

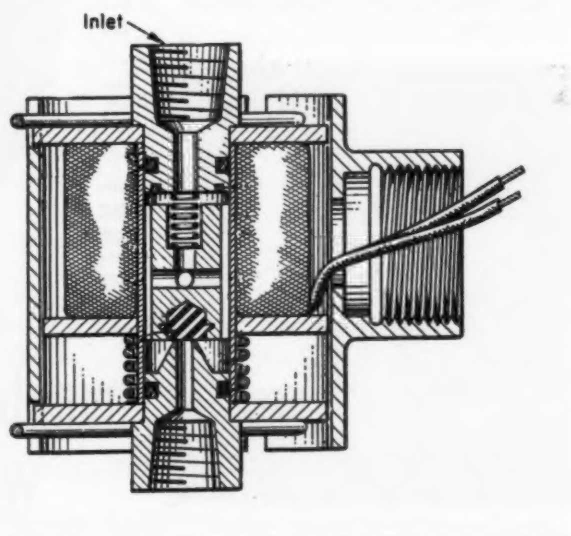
In bending roll service, an extremely high torque is required when the plate is first started through the rolls. In the Brakemotors used on a typical size of Niagara Bending Roll, a 7½ hp motor provides the same starting torque as a 15 hp motor. Moreover, this 7½ hp motor is built in a smaller than standard frame size.

These features of fast, accurate stopping, reversing and inching on short cycles and smaller frame sizes are made possible by the brake action which eliminates the need for plugging and by the added safety factors for heavy duty which are built into the motors.

The Brakemotors are built by Star-Kimble Motor Division of Miehle Printing Press and Mfg. Co., 201 Bloomfield Avenue, Bloomfield, N. J. Complete details are given in Bulletin B-501-A, copies of which are available on request from the manufacturer.

NOTEWORTHY
Patents

ELECTROMAGNETIC FLOW CONTROL for hydraulic systems is provided by the valve described in patent 2,616,452. Invented by Robert B. Clay and George A. Goepfrich, the valve is designed for normally-closed operation. Fluid flow through the valve is controlled by a sliding valve plunger actuated by an electromagnetic coil. When the coil is energized the



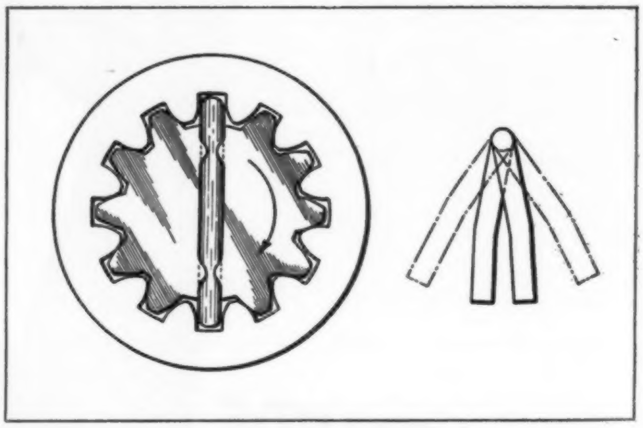
valve opens. Fluid flow through the valve acts to assist the valve closing action as well as providing lubrication for the valve plunger. Valve construction, which can be also adapted for normally-open operation, facilitates assembly and maintenance. The patent has been assigned to the Skinner Chuck Co.

VARIABLE TORQUE TRANSMISSION, both in direction and magnitude, is achieved in a novel magnetic-fluid clutch design which employs a planetary gearing arrangement. Covered in patent 2,616,539, the clutch has two rotating magnetic coils which are separately excited and positioned on opposite sides of a flat metallic plate connected to the output shaft. The system of planetary gears causes the coils to rotate in opposite directions. By varying the excitation current in the coils, torque and speed of the output shaft can be varied from a maximum value in one direction of rotation through zero to a maximum value in the opposite direction, while the input shaft continues to rotate in one direction at a constant speed. Output torque is a function of the difference in the

two exciting currents and the direction of rotation is determined by the maximum current. Invented by Vernon J. Wattenberger, the patent has been assigned to General Electric Co.

Laminated lock nuts for large machine bolts, built up like Belleville springs, are described in patent 2,616,474. Locking action is provided by dished metal disks, stamped from sheet stock and stacked in pairs, which flatten when threaded on the bolt and produce a binding pressure. The patent has been assigned to Harvey Hubbell Inc. by Joseph F. Healy.

Cushioned backlash takeup for splined power connections is provided by a unique device assigned to General Motors Corp. under patent 2,615,315 by Calvin J. Werner. Engagement of the splined "teeth" on the driving and driven members is maintained even during load reversals, eliminating the noise

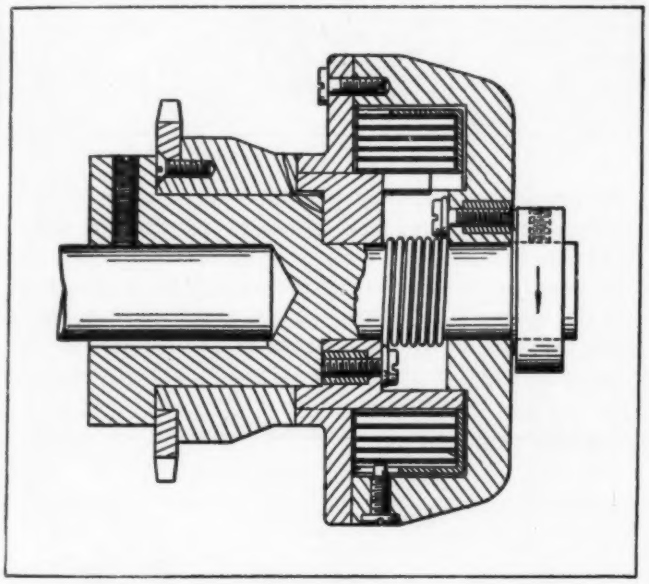


and destructive pounding which can occur when clearance has been provided between adjacent teeth. The torsional force necessary to hold the teeth in contact is supplied by a U-shaped spring rod fitted over the end of the drive shaft; two splines are removed to make slots for receiving the arms. Under impact loads the spring arms act as cushions to absorb the shock.

Automatic friction adjustment is afforded in a variable-speed friction drive which compensates for wear and torque variations. Assigned to Barnes Drill Co. under patent 2,617,309, the drive employs two offset friction disks, one flat and the other spherical, to obtain variations in output speeds. A unique cam arrangement on the drive shaft forces the disks into engagement and varies the contact pressure to provide a frictional force corresponding to the torque being transmitted. Irregularities in the friction surfaces as well as vibration are compensated by

a resilient mounting on the drive motor. Adjustment of output speed is accomplished by tilting the motor through a hydraulic-positioning control system. Inventor of the drive is Kenneth H. Casson.

OVERLOAD TORQUE PROTECTION, regardless of direction of rotation, is afforded by the clutch covered in patent 2,618,137. Assigned to Eastman Kodak Co. by William M. White, the clutch may be preset to automatically disconnect when a prescribed load value is exceeded. Under normal operating conditions, torque is transmitted by a coil spring engaging the drive shaft through a torque-limiting spring to the driven member. When the load becomes excessive, however, the torque-limiting spring permits the driven member to slip causing the coil spring to unwind and disengage the shaft. Adjustment of the tension in the torque-limiting spring, which may be accomplished without disassembly, determines the load at which release will occur.



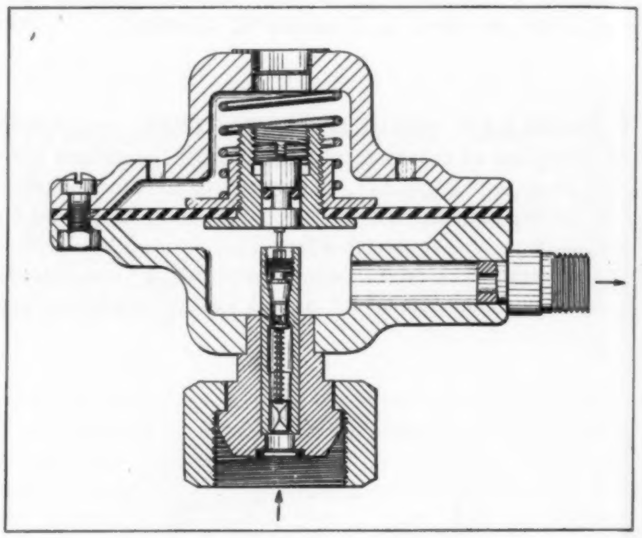
PRESSURE-RESPONSIVE SEAL in a union for hydraulic pipe or tubing is actuated by fluid pressure to provide more effective sealing. Assigned to Chiksan Co. under patent 2,610,870 by Daniel J. Parmesan, the union employs a resilient seal ring, trapezoidal in cross section, between coupling members. Sealing is effective even with slight misalignment and ribs on the resilient contact faces act to provide a labyrinth sealing effect.

END-BRACED ROTOR decreases rotary gear pump size and weight for a given capacity. Employed in

the pump of patent 2,615,399, the rotor may be employed in longer lengths than conventional types, with a resultant reduction in diameter and weight. Pumping action is provided by an internal idler gear offset from the rotor center. Bracing of the rotor is accomplished by an end support ring fastened to the rotor-bar ends by screws. The construction minimizes the possibility of rotor damage due to excessive pressure and reduces manufacturing costs. Assigned to Peerless Machinery Co., the patent is an invention of Milon Gay Edwards.

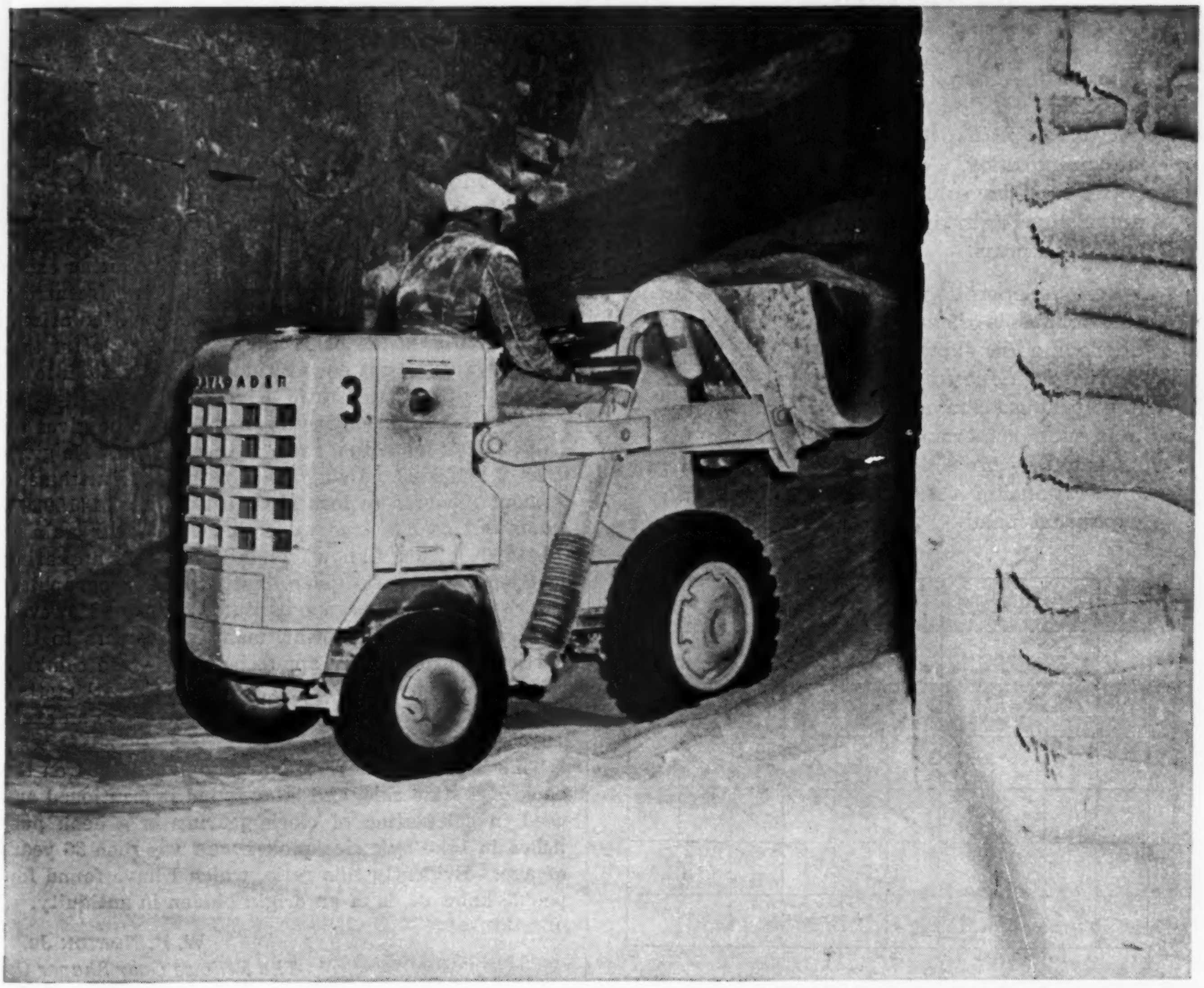
AUTOMATIC FLUID CUTOFF in hydraulic lines to prevent transmission of excessive pressures is effected with a fluid fuse invented by Allen J. Mellert and detailed under patent 2,615,675. Assigned to the Carpenter Mfg. Corp., the fuse is inserted into a hydraulic line at the desired control point. Excessive line pressure actuates a dished spring-metal diaphragm, which has a central opening for normal flow, causing it to flatten against a rubber sealing cup and cut off further flow. Restoration of the correct pressure differential returns the diaphragm to its normal flow position. The fuse can also be adapted for gas or vacuum operation.

GAS PRESSURE REGULATION is achieved with a conventional tire valve in the regulator assigned to Air Reduction Co. under patent 2,615,287. High-pressure gas enters the regulator chamber through the tire valve and is reduced to the predetermined deliv-



ery pressure. Delivery pressure is maintained by a spring - loaded, pressure - actuated diaphragm which controls the tire valve-stem position thus regulating the admittance of the high-pressure gas. Adjustment of the delivery pressure can be made during operation through an access opening in the top of the regulator. Inventor of the regulator is John S. Senesky.

U. S. Rubber Multi-Flex Boots make equipment last longer



Note Multi-Flex Boot extended from ram plunger that lifts 12 cu. ft. bucket. The ram plunger activates the bucket to any position required for loading or unloading stone, cinders, sand.

The hydraulic actuating pistons on many types of equipment such as the tractor-shovel (above) and lift trucks are protected by Multi-Flex Boots. Made without molds by a versatile process developed by United States Rubber Company, they can do many jobs conventional boots cannot do. Multi-Flex Boots prevent scoring of pistons and damage to packing by stones, sand, cinders and other materials. They also protect against rust and corrosion.

The versatility of rubber seems to have no limits when in the hands of "U.S." technicians. They may be able to make your product last longer, or operate more economically. Write to address below.

Here the Multi-Flex Boot is compressed. The drive wheels throw quantities of stones against the boot at high velocity.

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PROFESSIONAL

Viewpoints

"... using plain fifth-grade arithmetic ..."

To The Editor:

In the February issue of MACHINE DESIGN I have studied with interest the article on Gear Ratio Logarithms as presented by Mr. George M. Mencke of Wilmette, Ill.

The selection of proper gearing to give rather exact ratios has been for many years one of my recurrent duties. During that time the pi ratio has often arisen, and I have used the basic 355 over 113 ratio for this purpose, breaking it down as does Mr. Mencke into 71 and 113 with a 5 to 1 ratio in series.

I take exception, however, to the value he gives on Page 178 as the actual ratio of these gears. He gives it as 3.14158292. Using plain fifth-grade arithmetic without recourse to logs I get a ratio of 3.14159292, which is closer to true value than his 6-gear train at 3.1415900. This, I know from his statement appearing just below the ratio given, is due to a typographical error, an 8 having been substituted for a 9. However, as it appears in the print, many newcomers to this particular problem are apt to be misled into thinking that his 6-gear train is the better. A small matter, no doubt, but mathematics is supposed to be an exact science.

This 355 over 113 ratio I first found in 1904 on the back of a slide rule, and later found it mentioned and used in calculation of clock gearing in a book published in 1826 by a clockmaker who was then 80 years of age. Evidently this ratio, which I have found few people know of, is of an origin hidden in antiquity.

W. P. NORTON JR.
The Fellows Gear Shaper Co.
Springfield, Vt.

They Say...

"It is hard to find a modern manufacturing business whose products and manufacturing and operating methods are not the result of inventions. At least half of the capital of the U. S., other than land holdings, is invested in enterprises founded upon inventions made in the last 150 years. Large and small companies alike need the stimulus of new ideas, improvements in design and methods, and new fields and products so that they may continue to grow and keep pace with the times. Here is the place for the research men, the inventors, the engineers who recognize the needs of modern day living and on whose productivity depends the progress of the various industries."—HAL F. FRUTH, *consultant and physicist, H. F. Fruth & Associates.*

The *ELLIOTT* CROCKER-WHEELER *Brake* MOTOR

■The bonded metal brake linings of the C-W brake give greater friction for quicker stops, greater wear resistance, complete immunity to climatic conditions, oil or grease, and long service with no change in retarding torque.

The unusually short overhang of the C-W brake provides maximum rigidity and the compactness that adapts it to limited space — a vital feature on machine drives. The magnetic action which releases the brake is instantaneous and powerful. The entire mechanism is completely enclosed in an easily removable cover.

Here is a brake you can depend upon to do its job day in and day out with true Crocker-Wheeler reliability. Available as part of any C-W integral motor, or separately for mounting on NEMA D flange motors, frames 203-326, or C face motors, frames 364-405.

GET BRAKE MOTOR BULLETIN SL-610-1, addressing your request to Elliott Company Dept. MD, Jeannette, Pa.

ELLIOTT Company

CROCKER-WHEELER DIVISION
Ampere, N. J.

BRANCH OFFICES AND REPRESENTATIVES IN PRINCIPAL CITIES
ELLIOTT *Approved* SERVICE SHOPS COVER THE COUNTRY

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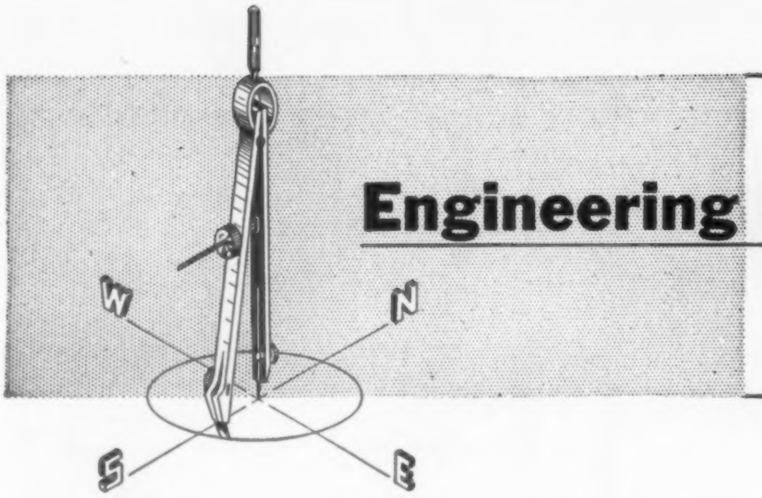
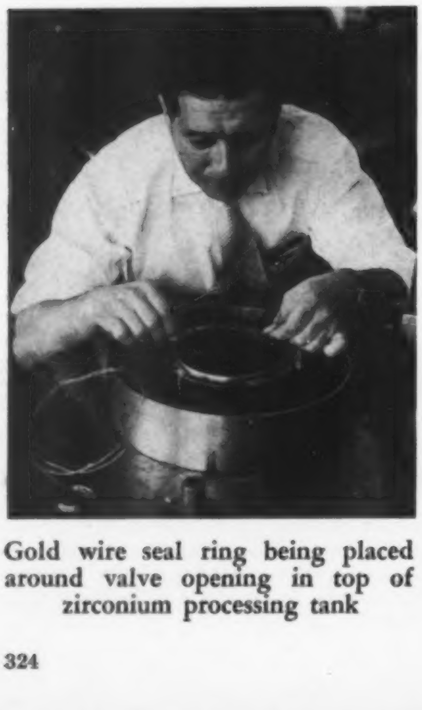


Engineering News Roundup

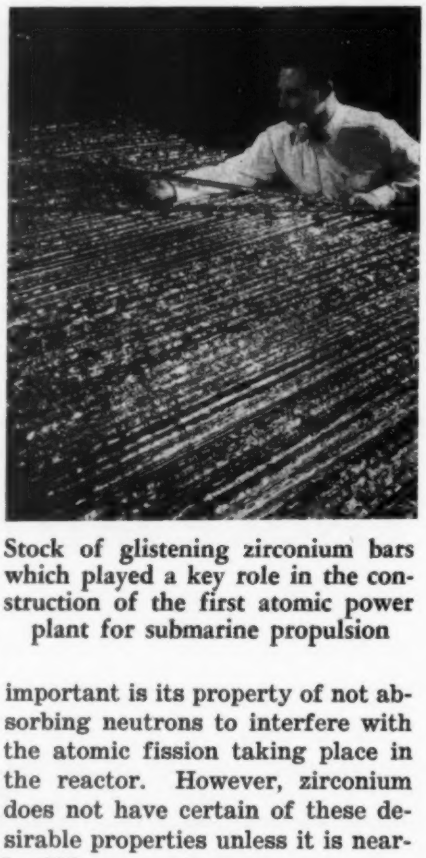
Golden Gaskets Help Produce Pure Zirconium

Pure gold wire is the cheapest satisfactory material available for sealing large metal caps and valves at high vacuums and temperatures required in the production of pure zirconium by the Westinghouse Electric Corp. Although gold costs $35 an ounce, the wire can be reprocessed and used over and over again, which cuts the cost to $1 per ounce.

Zirconium metal is lighter than steel, remarkably corrosion resistant and has an extremely high melting point. A fine structural metal, due to its strength and workability, it is one of the best materials which may be used for nuclear reactor construction. Most important is its property of not absorbing neutrons to interfere with the atomic fission taking place in the reactor. However, zirconium does not have certain of these desirable properties unless it is nearly 100 per cent pure.



Gold wire seal ring being placed around valve opening in top of zirconium processing tank



Stock of glistening zirconium bars which played a key role in the construction of the first atomic power plant for submarine propulsion

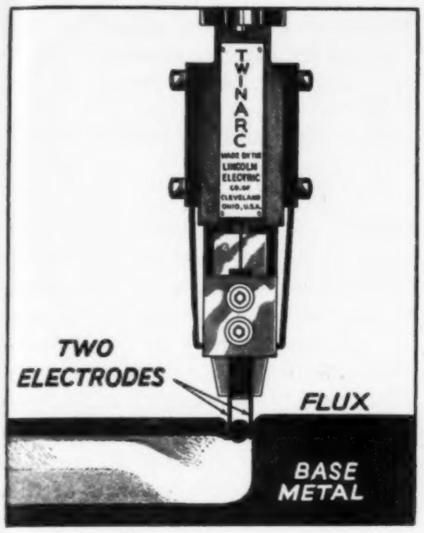
When Westinghouse scientists were given the job of building the first reactor for submarine use, enough usable zirconium was not available. The Atomic Energy Commission gave them permission to undertake mass production of zirconium, and in 14 weeks the Westinghouse scientists, headed by Dr. Z M. Shapiro and Alexander Squire, set up a zirconium refining plant and had it in full production. In those 14 weeks, production of pure zirconium bars was boosted from hundreds to thousands of pounds per month.

Group Will Study European Research Facilities

"American industry can procure research talent and facilities in Europe for a third to one-half of what they cost in this country," according to J. E. Hobson, director of Stanford Research Institute. "Scientific manpower here is not keeping up with research demands. We believe that many companies might benefit in the long run if a closer relationship could be established between European research capacities and American research needs," he said.

These statements were made on the occasion of Dr. Hobson's announcing plans for a tour by American research executives of technical and research centers in Italy, Switzerland and France with the idea of utilizing laboratories in those countries for research programs.

Stanford Research Institute's International Division has reported on various European scientific centers during the past two years. The forthcoming trip was organized by the division in response to requests by industrialists that a tour to these research facilities be made. The itinerary to be followed will include Paris, Rome, Naples, Florence, Turin, Milan, Venice and Zurich. In addition to research centers in the major cities, the group of about 30 research executives will visit universities in Rome, Naples and Padua and the polytechnics of Turin and Milan for discussions with leading Italian scientists.



Twin Electrodes Speed Automatic Welding

A new development in hidden or submerged arc welding is said to increase the speed of automatic welding by 50 per cent or more. In addition to increasing the speed of automatic welding, the process is said to produce high quality welds on rusty or dirty plate where normal techniques would result in pinholes. Developed by the Lincoln Electric Co. of Cleveland and known as Twinarc welding, the process employs two small electrodes side-by-side in place of one large electrode.

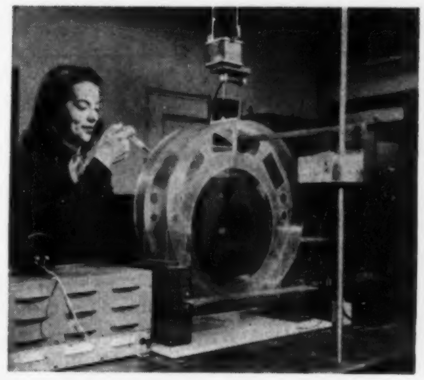
Both electrodes feed simultaneously through a single head and jaw to deposit metal in the weld crater. Higher currents possible with two electrodes increase the amount of penetration, rate of metal deposition and, consequently, welding speeds. Typical of the speeds possible is 15 inches per minute at 1500 amperes, used in making a ¾-inch fillet weld.

Materials Import Study Shows Areas of Weakness

A recent study by the Defense Production Administration shows materials that might be difficult to obtain if the U. S. were to be cut off from foreign sources of supply. It can provide a guide for long-range materials planning, but, at present, more than half the materials studied are not in short supply. Raw materials studied, along with percentage dependency on foreign sources and the present supply situation, are:

Antimony	75%	Ample
Asbestos	95%	Critical to moderately short
Bauxite (aluminum)	65%	Adequate
Beryl (beryllium)	90%	Very tight
Bismuth	50%	Adequate
Chromite (chromium)	99%	Demand exceeds supply
Cobalt	90%	Demand exceeds supply
Columbium	100%	Critically short
Copper	35%	Demand exceeds supply
Fluorspar (flux)	35%	No supply problem
Ilmenite (titanium)	32%	Adequate
Industrial diamonds	100%	Just meets demand
Iron ore	8%	Ample supply
Jewel bearings	90%	Adequate
Lead	45%	Adequate
Manganese	90%	Adequate
Mercury	90%	Adequate
Mica	95%	Barely meets demand
Nickel	99%	Critically short
Platinum group	90%	About in balance
Quartz crystals	100%	Adequate
Natural rubber	100%	Recent improvement
Rutile (titanium)	31%	In balance
Selenium	34%	Meets 80% of demand
Tin	100%	Demand exceeds supply
Tungsten	52%	Supply exceeds demand
Zinc	35%	Adequate

Copies of the complete report, *Raw Materials Imports: Area of Growing Dependency,* are published in two parts by the Office of Public Information, Defense Production Administration, Washington 25, D. C.



Plastic Models Used in Vibration Tests

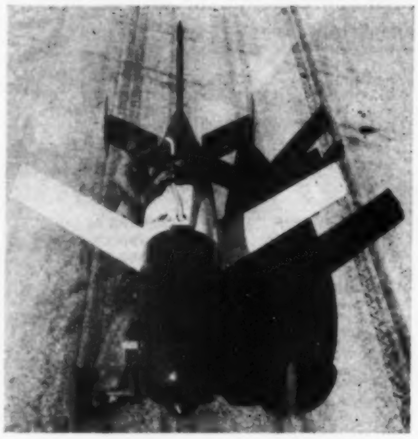
Plastic models are being used by Westinghouse Electric Corp. to determine effects of vibration on generators and other large electrical machines before the units are built. Used when obtaining such information by mathematical computation is impossible, the models save costly design changes in the actual apparatus.

In assembling the models, a hypodermic syringe is used to inject acetone and plastic cement into holes drilled through two parts to be joined. This method helps insure better fixing of sections of

BRAKE TESTER: Designed and built by Chrysler Corp. engineers, this laboratory machine is used to test the endurance of the company's power brake units. Normal braking, even from speeds up to 50 mph, seldom develops brake line pressures beyond a range of 160-260 psi; however, power brake units under test are made to generate pressures from 1000 to 1400 psi in the brake lines. Units must deliver at least 500,000 emergency "stops" without failure, and many complete 1.5 million maximum applications without a breakdown

the model which must withstand vibrations from the electromagnetic driver shown at the top of the plastic frame. Vibrations of a generator frame model are recorded with the aid of a crystal phonograph pickup. In testing, a varied range of vibration frequencies is applied by the electromagnetic driver, which is similar to the driving element of a radio loudspeaker.

Reveal Details Of Q-2 Jet Target Drone



Top view of Firebee, newly released by the Department of Defense, shows swept wing and tail, as well as air entry for its turbojet power plant

Somewhat less than half the size of present jet fighters, Ryan Aeronautical Company's Q-2 pilotless target drone displays performance characteristics comparable to jet aircraft now used in combat. The high-speed, high-altitude turbojet Firebee has been under test for two years, its principal use being that of a target for modern defense weapons. Simulating maneuvers of a piloted jet plane for training antiaircraft crews, it is equally adaptable for ground-to-air tracking and firing and for air-to-air interception problems.

Constructed of aluminum, magnesium and stainless steel, the Firebee is composed of five major assemblies: fuselage, nacelle, wing, empennage and parachute container. Wings, as well as tail assembly, attach to the fuselage with four readily accessible, self-aligning bolts. The nacelle containing the engine is hinged to permit ready access to the interior of the compartment.

Design includes a recovery system consisting of a two-stage parachute which decelerates the drone from near-sonic operating speed and lowers it to the ground without damaging the plane's structure or electronic controls. The parachute mechanism operates automatically in the event of a target

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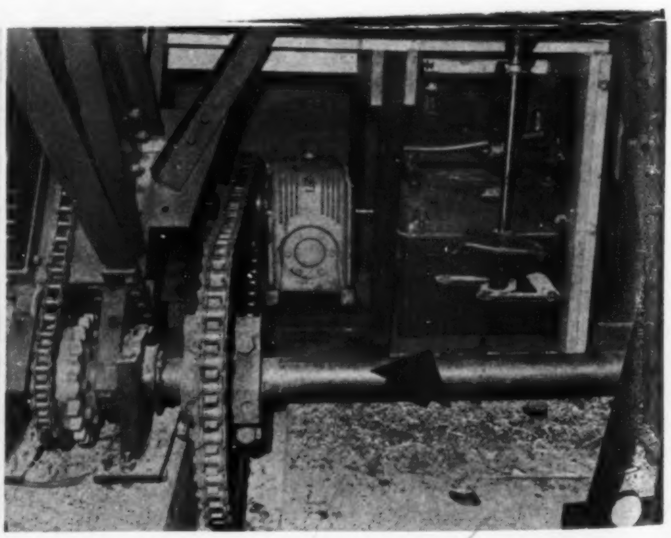
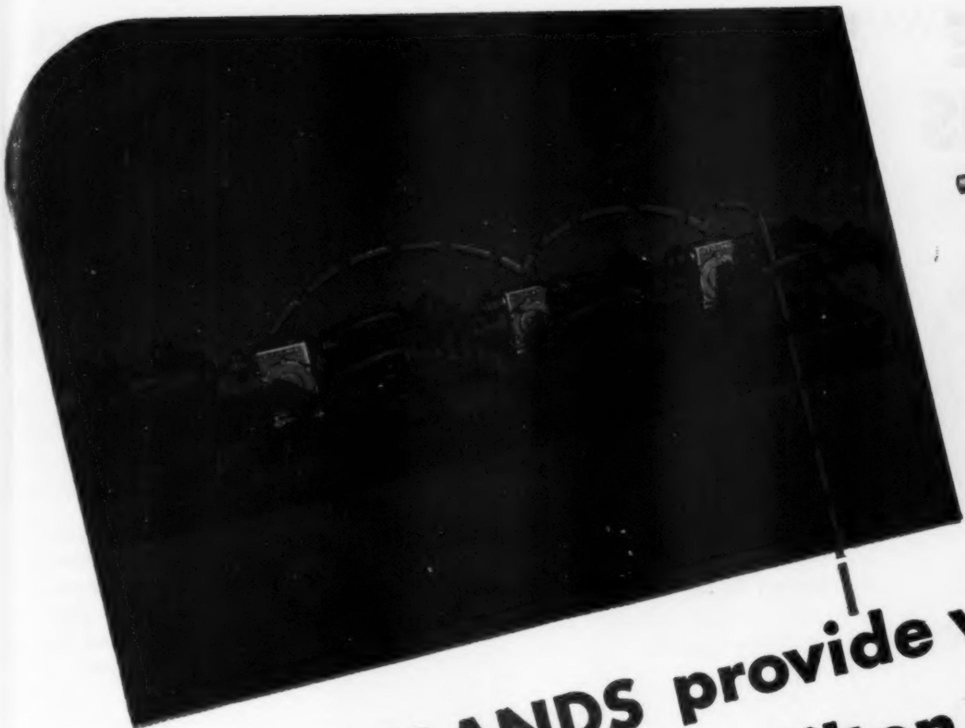
Lumber Unstacker Has High Capacity

A novel machine for unstacking lumber from the drying kiln cars is estimated to have a daily capacity of 200,000 ft. Until this machine was developed, it was necessary that two crews work overtime to unload and process the dried lumber at the Mt. Emily Lumber Co., Lagrande, Ore.

To unload dried lumber from a kiln car, the load is winched from the cooling shed to rails on the main deck of the unstacker. When the load is properly positioned, the entire unstacker tilts toward an apron leading to the grading table. As tilting progresses, the load is also hoisted or elevated so that it nearly clears the side frame of the unstacker by the time it is fully tilted. Hoisting continues, and as each course of lumber clears the side frame, the course slides by gravity onto the apron leading to the grading tables. Stickers, the separators between courses, slide end-wise under the apron onto another apron leading to an endless belt which carries them to the empty kiln cars of the previous load.

Control of both tilting and lifting speeds of the unstacker is provided by a Reeves variable speed transmission. Lifting speed is usually about one foot per minute while the return speed is five to six times the lifting rate. A large part in the development of the machine was played by Pacific Coast Geared Products Inc., Reeves Pulley Co. representatives in Portland, Ore.





Three Cleveland 10 RT speed reducers (with 3" centers) on a six cylinder gang pump installed in a California oil plant. Two of the worm gear units drive pumps at 88 strokes per minute, the third at 70 strokes per minute. Photo by courtesy of Milton Roy Company.

Small CLEVELANDS provide wide range of stroking speeds for Milton Roy pumps

THIS Milton Roy controlled volume pump meters 3 liquids in exact ratio—fatty acid, sulphuric acid, and glycerin, delivered at differing speeds.

To provide the variation of stroking speeds necessary and to insure accurate control of proportioning and volume, the pump builder uses three small Cleveland worm gear reducers with a variable speed motor.

Cleveland worm gear units take up a minimum of space. They transmit power with a smooth, uniform torque flow. Case-hardened steel worms mesh with nickel-bronze gears for long, hard, dependable service without appreciable wear.

Throughout the chemical and allied industries, you'll find Clevelands—big and small—driving pumps, agitators, mixers—in fact, every kind of equipment for which a right-angle drive is needed.

If you have a power transmission problem, you can get help by writing for Catalog 400. The Cleveland Worm & Gear Company, 3265 East 80th St., Cleveland 4, Ohio.

Affiliate: The Farval Corporation, Centralized Systems of Lubrication. In Canada: Peacock Brothers Limited.

CLEVELAND Worm Gear Speed Reducers

hit, loss of radio-wave carrier from its remote control station, engine failure, exhaustion of fuel supply or at the direction of the remote control operator. The parachute detaches from the drone when contact is made with the ground, thus preventing damage due to dragging by ground winds.

The Firebee is powered by a Fairchild J-44 turbojet engine which develops about 950 pounds thrust and is approximately 6 feet long and 22 inches in diameter. Wings and tail surfaces are sharply swept back. The craft has a wingspan of approximately 12 feet and is 18 feet long. It weighs about 1800 pounds.

Firebee in flight during tests in New Mexico. Controlled by radio, the pilotless drone can be flown at high altitudes and out of sight of its remote-control operator

New Powder Metal Parts Have Improved Ductility

A new iron powder metal having improved ability to withstand high stresses without cracking can be used to make finished machine parts such as gears, cams, brackets, and lever arms. Such parts previously had to be made from forgings or bar stock due to inadequate strength of powder metals. The new product, developed by Amplex Div., Chrysler Corp., is called Steel Oilite.

Physical properties are said to be comparable to those of mild carbon steels such as SAE 1010, 1020 or 1030. Regular powder iron products will withstand pressures of approximately 35,000 psi; the new product will withstand twice that. Ductility of the new metal is two to three times that of other powder metal parts produced by Amplex. Though the material is porous it is not intended for self-lubricating applications.

Produced by pressing, sintering and finish sizing in a press, the parts may be plated by any of the normal processes. Steel Oilite may also be hardened by direct quenching or carburized and hardened.

More Steel per Person Than Any Other Country

More steel per person was used in the United States in 1951 than in any other country. Total per capita consumption in the U. S. was about 1347 pounds. Canada, with 805 pounds per capita, was second.

Since before the war (average of

CRYSTAL BALL: The young lady behind the crystal ball is looking into the future of submarine propulsion. This plastic sphere is a model of a 225-foot steel ball now under construction by General Electric's Knolls Atomic Power Laboratory to house a land-based prototype of an atomic power plant for submarines. The sphere will contain a complete submarine hull section, as shown inside the plastic ball. Part of the hull will be surrounded by water, making possible testing under conditions closely approximating those encountered at sea

(Page missing)

1936 to 1938) average consumption has increased 646 pounds from the usage of 701 pounds during these years. Estimated capacity of 117.5 million tons available at the beginning of 1953 is 1483 pounds per capita.

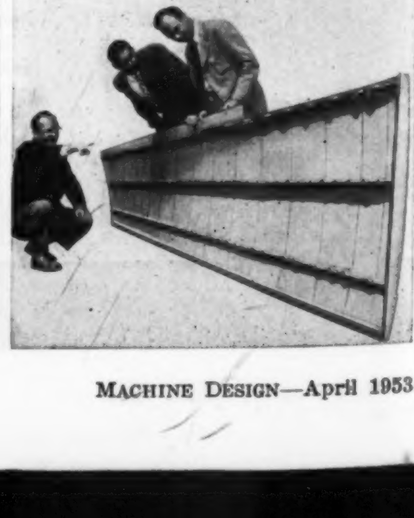
In third place was Sweden with 710 pounds, followed by Australia with 633 pounds; United Kingdom, 611 pounds; Belgium-Luxemburg, 553; Germany, 483; and France, 410. Russia's consumption was about 342 pounds.

Record Motor Production Predicted for Next Decade

More electric motors will be built and sold between now and 1963 than have been manufactured in the past 50 years, according to F. C. Ruling, manager of General Electric's Atlantic district. Since the market for industrial products is expanding more rapidly than ever before, the electrical manufacturing industry is expected to grow twice as fast as the rest of the economy in the coming years, he said.

•

CAST MAGNESIUM WING: Believed to be the largest cast aircraft surface ever produced, this 16-foot, one-piece magnesium wing section was developed by Northrop Aircraft Inc. and the Aluminum Co. of America. Such wings, fabricated from AZ-92 magnesium alloy, can be produced more rapidly and economically than wings made by spot welding or riveting aluminum skin to the spars and ribs, according to Alcoa. Although the magnesium wing section is only ¼-inch thick in some spots, it is said to conform fully to aircraft quality standards



Plating Processes Require No Nickel

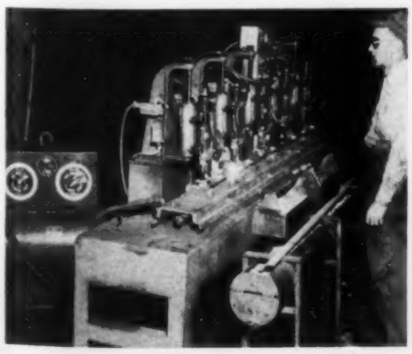
Two decorative plating processes, for use on parts commonly having a copper-nickel-chrome finish, have been developed. They require no nickel, can be readily deposited over steel or copper, and are readily chrome-plated. Brilliance is reported to be comparable to copper-nickel-chrome. Protection of steel against red rust in a standard salt spray test is claimed to be as high as 200 hours, or twice the test life of good quality copper-nickel-chrome.

Designated Probite CR-723, the finishes are a buffing-grade deposit for parts that can be color buffed after plating, and a sealer for use over the chrome plating. Complete information may be obtained from Promat Div., Poor and Co., 851 S. Market St., Waukegan, Ill., developers of these finishes.

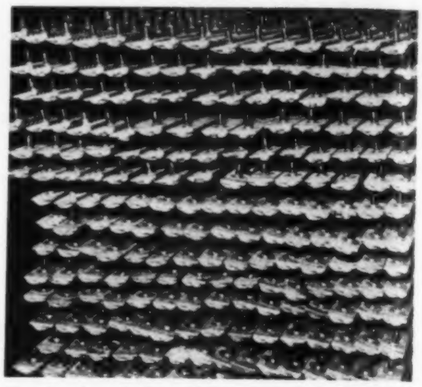
Welded Studs Speed Harvester Assembly

A special five-gun stud welding unit in use at the East Moline, Illinois plant of the International Harvester Co. welds 10 studs to a harvester-thresher rasp bar in less than 40 seconds. The complete cycle includes loading studs into the welding guns, positioning the work and welding ten studs five at a time by shifting the rasp bar seven inches with an indexing device. The granular flux-filled studs and stud welding unit are produced by Nelson Stud Welding Div. of

Stud welder attaches ten studs to the rasp bar in less than 40 seconds



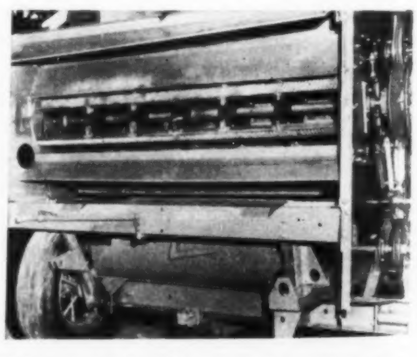
Engineering News



Rack loaded with completed bars

Gregory Industries Inc.

Through-bolts were previously used on the rasp bars, which were drilled and countersunk so the bolt-heads would not project beyond the stripping surface. Stud welding has eliminated the danger that bolts made to conform to the contour face would turn to positions that would interfere with smooth operation. Some implement manufacturers using both single and multiple stud-welding units report savings up to $1.54 per combine cylinder assembly. Field removal and replacement have also been simplified as nuts holding the rasp bars to the backup plates can be removed from the studs although the nuts may be rusted or fouled.



Harvester-thresher unit showing rasp bar cylinder assembly in place

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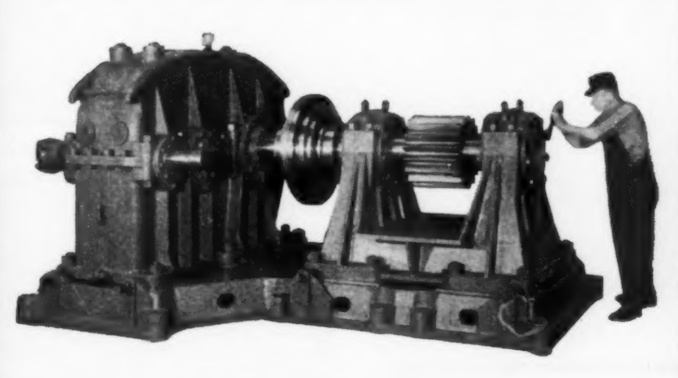
James W. Murphy was recently named manager of sales of the stainless and alloy castings division of **Allegheny Ludlum Steel Corp.** at Buffalo. Mr. Murphy has been associated with the company since 1936 and has served as assistant sales manager of the castings division for several months.

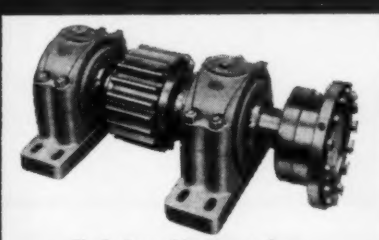
When It Comes to
POWER TRANSMISSION
On the BIG JOBS
Look to... Jones



Cement Kiln Drive

JONES Products—*While specifically* designed for the heavy industries — *are competitive for all applications.* All the added values of sturdier design, heavier construction and better materials are yours without extra cost.

Ask for Bulletins



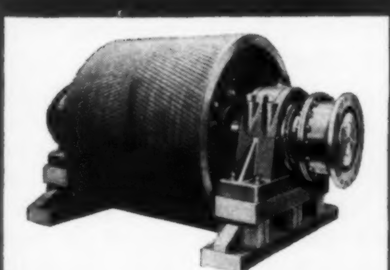
Shaft Assembly Mounted on 4-15/16" Pillow Blocks

SPEED REDUCERS AND COMPLETE DRIVES TO 500 HORSEPOWER

HERRINGBONE   WORM GEAR   WORM HELICAL

PILLOW BLOCKS to 9" Shaft Size

Timken Equipped, Adaptor Sleeve Type
Self-Aligning, Dust-Tight
Sizes to 4-15/16" in Stock



Jones 60" x 57" Rubber Covered Pulley mounted on 9" Jones Timken Pillow Blocks

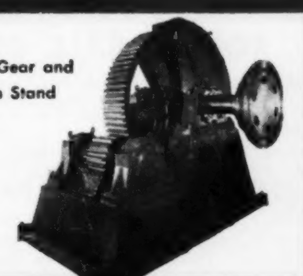
CAST IRON CONVEYOR PULLEYS

Tailor-Made to Your Requirements.

CUT GEARS

to Your Specifications

HERRINGBONE - WORM
HELICAL - BEVEL - SPUR



Spur Gear and Pinion Stand

W. A. JONES FOUNDRY & MACHINE CO., 4413 W. Roosevelt Rd., Chicago 24, Ill.

Since 1890

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HERRINGBONE — WORM — SPUR — GEAR SPEED REDUCERS
CAST IRON PULLEYS • CUT GEARS • V-BELT SHEAVES
ANTI-FRICTION PILLOW BLOCKS • FLEXIBLE COUPLINGS

In The Service Of Industry

Fluid Power Controls To Be Subject of MIT Course

Planned especially for engineers whose work is in some phase of the fluid power controls field, an advanced course in this subject will be conducted by members of the Mechanical Engineering Department of Massachusetts Institute of Technology in co-operation with the staff of the Dynamic Analysis and Control Laboratory. Professor John A. Hrones, director of this laboratory, will be in charge of the two weeks' program.

Highlights of the course will include discussions of the fundamentals of fluid flow, new concepts of flow valve performance and design, and the generation and utilization of compressible and incompressible fluid power. Analytical and simulation techniques for studying the dynamics of valve controlled systems will be discussed, and there will be appropriate laboratory demonstrations.

The course will be given July 6-17, with enrollment limited to those whose industrial and educational experience will enable them to contribute to and benefit from the program. Further information and application blanks may be obtained from the Director of the Summer Session, Room 3-107. Cambridge 39, Mass.

♦

An addition of 25,000 sq ft of manufacturing space is being planned by the **American Welding & Mfg. Co.,** Warren, O. The second step in a $5 million expansion program, this addition will consist of a 20,000-sq ft structure to house machine tool equipment. First step in the expansion program was a new building opened for production in March 1952. In addition, present production area of the company's Warren Machine & Die Div. will be doubled.

♦

A new laboratory which provides complete facilities for testing and developing materials, equipment and processes for customers has been opened at the **F. J. Stokes Machine Co.,** Philadelphia.

Mechanical Engineers Needed in Electronics Field

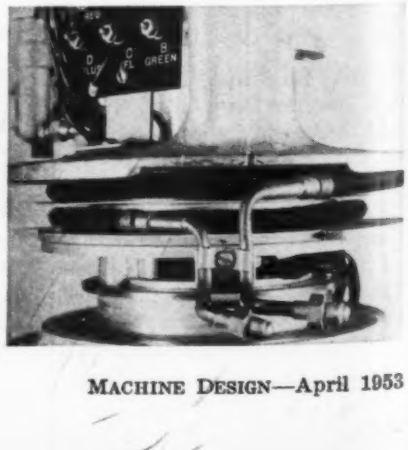
Mechanical engineers have a bright future in the field of electronics, acording to A.C. DeNapoli Jr., chief mechanical engineer of Motorola Inc. In an address entitled "The Mechanical Engineer: His Future Possibilities in the Electronics Industry," given before the Chicago chapter of ASME, he traced the growth of the electronics industry, pointing out that the development of sound motion pictures was the first important collaboration of electronic and mechanical engineers.

With the growth of radio, television and military and industrial electronics, the importance of good mechanical engineering as well as circuitry has brought the mechanical engineer closer to the electronics field, according to Mr. DeNapoli. He predicted that, although many present mechanical devices will be operated electronically in the future, this electronic growth will actually increase, rather than diminish, the need for mechanical engineers.

Metallic Hose Replaces Swivel Joints

Gas lines to the bunsen burner and hot and cold air lines made of Titeflex convoluted flexible metal tubing eliminate the need of ball or swivel joints in the S. S. White Dental Company's Master Dental surgery room unit. The 90-degree swing motion of the accessory table arm of the unit is taken up by the tubing, which flexes gently around the arm bearing column.

Each flexible section is one and



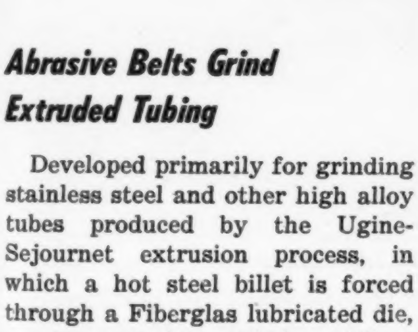
one-half turns around the column. As the arm moves the flexible tubing slackens and rides away from the column or the slack is taken up depending on the direction of rotation of the arm. Ends of the flexible sections are connected to rigid seamless tube lines at both supply and table arm ends. All joints are, therefore, secured against movement and the tendency to develop leaks.

The bronze, single braid hose used has a special braided cotton cover to eliminate possible noise. As flexing of the hose in more than one plane is not recommended by the manufacturer, the flat supports shown in the accompanying photograph were installed to confine all flexing to the horizontal plane.

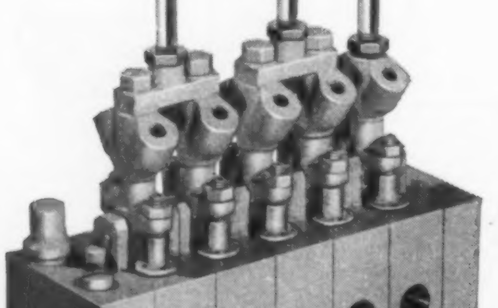
To allow maximum space for flexing of the hoselines, all electrical lead wires are installed in the center column of the stand.

Abrasive Belts Grind Extruded Tubing

Developed primarily for grinding stainless steel and other high alloy tubes produced by the Ugine-Sejournet extrusion process, in which a hot steel billet is forced through a Fiberglas lubricated die, a new centerless grinder utilizes two coated abrasive belts. Two pounds of stock can be removed from a 15-foot length of 2.375-inch OD tubing in two minutes. The result of co-operative engineering between Behr-Manning Corp. and Production Machine Co., the new process has as its objectives (1) removal of stock from the extruded tubing at a faster rate than obtainable by any other method of



Our business



ALL-HYDRAULIC TraveLoader

Latest advance in self-propelled, high-speed loader uses Blackhawk Hydraulics throughout. Three sizes of rams, V-283V VALVBANK and P-104 Pump provide adjustment of conveyor for varying truck heights . . . raising of front wheels for highway travel . . . lifting of feeder for cleaning and servicing.

For low-cost, positive-sealing control valves — standard VALVBANK sections can be combined in unlimited combinations

Blackhawk VALVBANKS are without comparison for controlling the flow of oil into single- or double-acting rams or any other hydraulic device requiring corresponding control. The V-28 valve (shown at left in a VALVBANK) is an open-center, series type, precision-control valve for all pressure ranges. Hydraulically balanced, it handles up to 6000 psi working pressures with finger-tip actuation. Flow capacity up to 6 gpm. VALVBANKS can be built up in any combination of single- and double-acting control units and include a relief valve. Spring-return to neutral "hold" position provides "dead-man control." All ports are ½" standard standard pipe thread.

is HYDRAULICS...
to help "power" your business

Blackhawk is a creative source of supply for hundreds of successful manufacturers

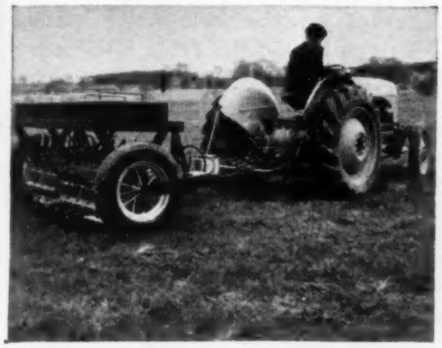
If you build agricultural, construction, or materials handling machinery — *or any other* product requiring positive, long-life, low-cost control — it will pay you to investigate the sales-building advantages of Blackhawk Hydraulics. For a single component, a complete hydraulic system or a self-contained hydraulic tool — here are three of many reasons why Blackhawk can help you build new performance, new economies into your product:

1. LOWER COST PER COMPONENT — Continuous, full-scale line production brings startling savings on a broad range of standard units.

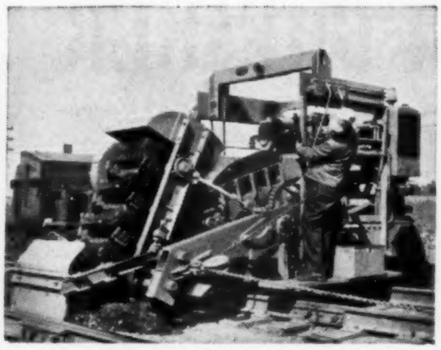
2. SAVINGS IN OTHER MATERIALS AND IN ASSEMBLY TIME — Dramatically compact design saves space and weight. Components for systems in all pressure ranges.

3. CUSTOMER SATISFACTION — Simplified construction has proved itself to be long-lived under the most grueling field conditions.

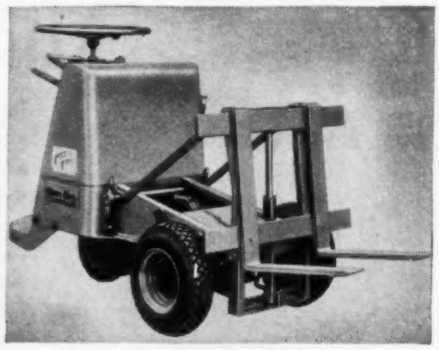
You are invited to take full advantage of Blackhawk's unmatched engineering and manufacturing facilities. Write for detailed engineering literature on the complete line of Blackhawk pumps, rams and valves for original equipment applications.



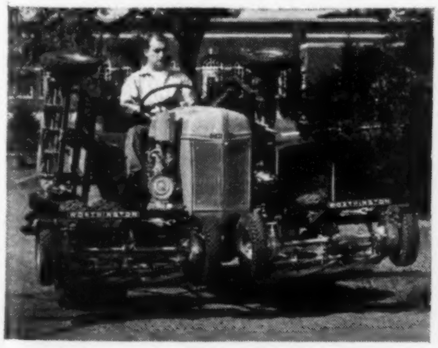
DIVOT DIGGER — Turf aerator loosens soil under grass cover without disturbing turf. Blackhawk hydraulic pump lifts and lowers the reel to clear obstacles.



BALLAST WASTER — Fouled railroad ballast is removed from roadbed and wasted over the bank. Blackhawk pump and valve control chain propulsion mechanism.



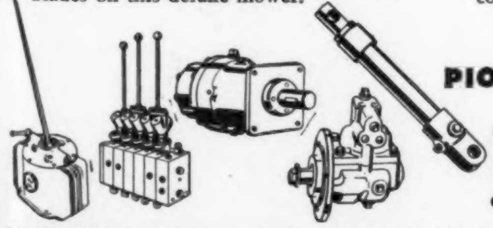
POWER WHEELBARROW can also be equipped with a variety of front-end attachments like this fork-lift which uses Blackhawk hydraulics to lift 500 lbs.



TRACTOR-MOWER — Blackhawk pump, rams and valve provide positive raising and lowering of all five of the cutting blades on this deluxe mower.

SEEDLING PLANTER — Plow and disc assembly are raised and lowered by a Blackhawk ram and "Power-Packer,"® a self-contained pump, valve and reservoir unit.

SERVICE TRUCK — Blackhawk hydraulic hand-operated pump permits easy lifting of loads up to 6000 lbs. Hydraulic safety valve prevents dangerous overloads.



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BLACKHAWK MFG. CO., HYDRAULIC CONTROL DIVISION • DEPT. J-5443 • MILWAUKEE 1, WISCONSIN

grinding; (2) removal of the Fiberglas skin and extrusion die marks; (3) removal of taper from the tubing; and (4) imparting of an acceptable finish to the tubing.

A 50 grit aluminum oxide waterproof cloth abrasive belt 9 inches wide and 168 inches long does the grinding. Work is fed past the grinding belt by a 220 grit aluminum oxide waterproof cloth abrasive regulator belt 9¾ inches wide and 58 inches long. Lubrication is provided by a cutting oil which is sprayed on the belt by a fan type nozzle. The recirculating lubricant flushes away the grinding swarf and prevents rewelding of the chips.

A 20-inch diameter steel contact wheel mounted on a 4-inch shaft supports the grinding belt against the work. This wheel also serves as the belt drive pulley. Driven by V-belts from a 25-horsepower motor, it forces the belt past the work at surface speeds of 3500-10,000 fpm.

The entire regulator belt assembly can be adjusted about a horizontal axis to provide a work feed angle of from 0 to 15 degrees in either direction from the vertical, which, together with a variable speed drive, makes possible an infinitely variable workpiece feed rate from 0 to 30 fpm. Work feed angle of the coated abrasive regulator belt can be changed at will.

Uniform rate of stock removal is maintained across belt width by setting a small taper between the contact wheel spindle and the face of the back-up platen. As a result of uniform pressure across the face of the belt, dimensional accuracy is assured during belt life.

Engineering News

Extrude Large Tubing By Ugine-Sejournet Process

A new extrusion unit for producing stainless steel seamless tubing up to 6¼ inches in diameter has been placed in service at United States Steel Corporation's National Tube Div. plant in Gary, Ind. Maximum size with previous equipment was 5⅛ inches outside diameter. The new unit, employing the French Ugine-Sejournet process in which glass in various forms acts as a high-temperature lubricant for extrusion dies, consists of a 2500-ton capacity hydraulic press and related facilities.

Tubing can be produced from nonpierceable grades of stainless steel with the new facilities. Unusual shapes that cannot be rolled because of their unbalanced design can also be extruded.

The main press ram is powered by a 2000 ton hydraulic cylinder, and the piercing ram or mandrel by one of 500 tons. Power is transmitted through a closed hydraulic system using a solution of city water and soluble oil as fluid. Total fluid capacity is 8000 gallons, provided by a 1200-gallon prefill tank and a 6800-gallon gravity tank. Two hydropneumatic accumulator systems provide fluid pressure of 3900 psi. Maximum operating pressure requirement is 3600 psi. Air for charging and recharging the accumulators is provided by a 4000-psi compressor.

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WITH TRANSMISSION BRAKE

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For Cars • Trucks • Tractors • Farm Implements • Road Machinery • Aircraft • Tanks • Busses and Industrial Equipment

Engineering News

Frozen Aluminum Speeds Aircraft Production

A 40 by 12 foot refrigerator standing 11 feet high was placed in operation recently at the Burbank plant of Lockheed Aircraft Corp. Capable of cooling 2500 pounds of aluminum from room temperature to −20 F in two hours, the 5000 cubic foot deep freeze keeps plane parts soft between steps in fabrication.

Aluminum parts become soft during the heat-treating process which finally gives them high-strength and hardness. If allowed to cool at normal room temperature, they harden rapidly. Such hardness hinders further forming and shaping operations, but it has been found that quick freezing keeps the metal soft indefinitely. Parts heated to as much as 970 F in giant ovens are now quenched, rushed into the ice box and frozen until machines and men are ready to perform further operations.

Radioactive Source for Industrial Investigations

The largest radioactive source outside of the Atomic Energy Commission installations will become the heart of a new Radiation Engineering Laboratory at Stanford Research Institute. To be used in the investigation of industrial applications for nuclear energy—including the nondestructive testing of complicated metal castings and parts—a quantity of Cobalt 60, a strong gamma-ray emitter, was transferred to a laboratory tank containing 5400 gallons of water, which provides a totally safe shield.

The power source comprises four cylinders and a rod, each a foot in length, which can be nested or used in various combinations. The five elements weigh ten pounds when combined and are separately rated at 1480, 1200, 1100, 680 and 110 curies.

Gamma rays emitted from the source are similar in type to X-rays, although they have a

(*Continued on Page* 353)

(*Continued from Page* 248)
shorter wavelength and much greater penetration power than most commercial X-ray machines. Emissions from a source of the size of the new installation are stronger than a one million-volt X-ray machine.

Service will be provided companies wishing to explore uses of radiation for their processes or products. Samples may be brought to the Institute for irradiation at specified intensity and duration.

Largest Circuit Breakers Ordered for Atomic Project

Eighteen of the world's largest circuit breakers, each with an interrupting rating of 25 million kva at 330 kv, will be built by Westinghouse Electric Corp. for the Portsmouth, O., Atomic Energy Commission project.

Operating in connection with sensitive relaying equipment, the breakers will automatically stop short circuits or faults within 1/20 of a second. Within 1/3 of a second, when desired, the breaker will close again, re-establishing service if the fault has been removed.

♦

The **Tocco Industrial Heating Div.** of the **Ohio Crankshaft Co.,** Cleveland, has announced winners of its 1952 "Economy in Production" contest. Winner of the $1000 first prize was Lloyd E. Raymond, Metallurgist at Singer Mfg. Co., Bridgeport, Conn., whose paper showed how the Tocco process had reduced hardening time 62 per cent in the manufacture of sewing machine parts. Other prize winners are John P. Isaacs, Norris Thermador Corp., Los Angeles, and Henry C. Fischer, Army Chemical Center, Edgewood, Md., both of whom won $600; R. H. Lauderdale, Northern Ordnance Inc., Minneapolis, who was awarded $400; and J. C. Scullin, Cardinal Machine Co., Glendale, Calif., who won $100.

♦

A new 50,000-sq ft plant for the production of relays was recently completed by **C. P. Clare & Co.** in Chicago.

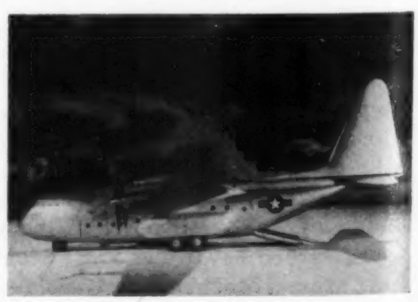
Develop New Insulation

A new insulation against heat, with potential applications in fields where heat and limited insulation space are factors, is under development at the General Electric research laboratory. According to research director C. G. Suits, the new insulation is ten times as efficient as any now in use. This factor made possible the use of half-inch thick walls in the prototype "XR-10" refrigerator-freezer, recently unveiled, in which the new insulation was used. Dr. Suits indicated that use of this insulation would provide extra refrigerated space and permit placing the refrigerator above the kitchen work counter rather than in an upright floor cabinet.

Lockheed To Produce Turbo-Jet Transports

Using new turboprop engines which harness jet power to conventional propellers, Lockheed's C-130 will fly faster and higher than any present military transport, according to its designers. Its four engines will have three-bladed propellers. Wingspan will be 132 feet; length, 95 feet; height, 38 feet. It will require only short takeoff and landing runs; with special tandem-wheel tricycle landing gear it will be able to operate off small emergency landing fields or on rough, unfinished air strips.

As shown in the accompanying photograph of a model, a built-in



loading ramp serves as the transport's rear door. This door can be lowered to truck bed level for straight-in loading or dropped to the ground and used as a ramp

for driving vehicles directly into the airplane. In addition, a large forward cargo door is provided for simultaneous front and rear loading. The rear door can be opened in flight to drop troops and equipment.

As a cargo plane, the C-130 will carry large pieces of equipment such as a 155-mm howitzer and a high-speed tractor. It can also be fitted as a hospital plane to carry patients on litters. The plane is described by Hall L. Hibbard, Lockheed vice president of engineering, as "a simple, rugged airplane" which will operate more economically than existing military transports and "perform a variety of air jobs."

♦

United States Rubber Co., New York, plans to build a new research center devoted to basic research in the fields of rubber, chemicals, synthetics, textiles and plastics. To be built in Emerson, N. J., the new center will initially house fundamental research activities. The company will continue research activities at its Passaic laboratories.

♦

A government-owned plant is being built and will be operated by **Chrysler Corp.** for modification and final processing of military tanks for Army Ordnance. The new plant will be located adjacent to manufacturing operations of the Chrysler tank plant in Newark, Del., and will function as an integral part of that facility.

♦

A $10 million machinery and plant expansion program is nearing completion at **Standard Pressed Steel Co.,** Jenkintown, Pa. When reconstruction has been completed, more than 650,000 sq ft of floor space will be under a single roof.

♦

The **Skinner Chuck Co.** has recently completed a new plant in New Britain, Conn., which provides complete facilities for the company and its electric valve division. It comprises 85,000 sq ft of floor space and has modern facilities for the production of lathe chucks and machine vises for machine tools and solenoid-operated valves.

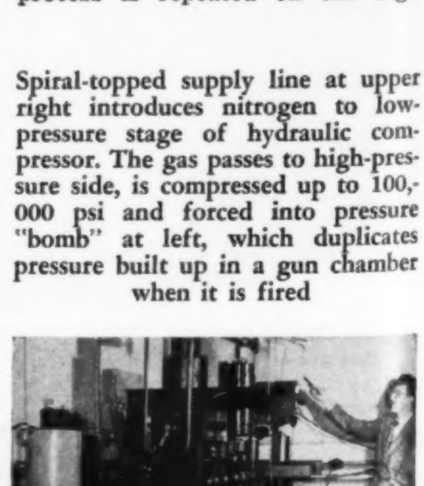
Engineering News

Automatic Compressor Facilitates Explosives Study

Capable of compressing gas to 100,000 psi, an automatic hydraulic compressor has been installed to facilitate study of the behavior of fuel burning at gun chamber pressures at the Naval Ordnance Laboratory.

The unusual compressor uses "liquid pistons" and has automatic hydraulic control and driving system. "Liquid pistons" eliminate the need for gas seals at the intensifier pistons. The gas does not come directly into contact with the compressing piston, but is compressed and confined by a liquid which is compressed by the hydraulically driven intensifier piston in another chamber. Nitrogen is being used presently in the compressor, but almost any gas except hydrogen can be utilized. Beginning in the low-pressure stage, the compression process uses nitrogen supplied from regular gas-storage cylinders. Hydraulic oil driven by the electric pump acts on a steel intensifier piston which displaces the special fluid into a "liquid piston" chamber where the actual gas compression takes place. When pressure reaches 15,000 psi, a valve opens, permitting hydraulic oil to pass into the high pressure stage of the compressor. The foregoing process is repeated on the high

Spiral-topped supply line at upper right introduces nitrogen to low-pressure stage of hydraulic compressor. The gas passes to high-pressure side, is compressed up to 100,000 psi and forced into pressure "bomb" at left, which duplicates pressure built up in a gun chamber when it is fired



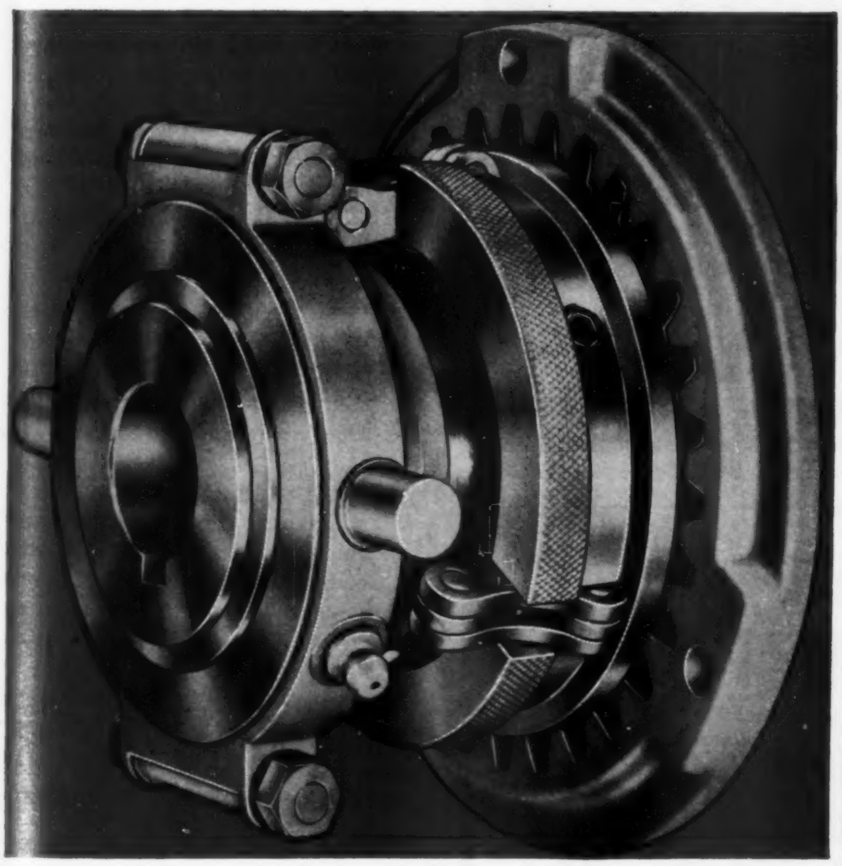
ANNOUNCING.....

THE NEW LOW COST

single disc

JOHNSON CLUTCHES

an addition to the* MAXITORQ *line



The Johnson 350 and 450 Single Disc Clutches are the newest additions to the Carlyle Johnson line... fitting companions to the Maxitorq multi-disc series.

They are ideal for light machinery service to 6 H.P. Several driving combinations are available, including V-belt. Far greater capacity at low cost is provided. (See column at right for typical applications.) They have the same "floating disc" principle as the Maxitorq Clutch ... discs that ride free in neutral ... no drag, no abrasion, no heating. A simple hex-key frees the knurled ring for easy manual clutch adjustment. Machine designers will find the solution to many problems with this new Johnson Clutch.

Send for Bulletin
MD-4-250

ICJ83

Frankly

SPEAKING

Because of a large backlog of orders, The Carlyle Johnson Machine Company found it necessary to postpone the announcement of the new Johnson Single Disc Clutches.

Now, with added equipment for high speed production, it is possible to produce and ship the new clutches without undue delay. In fact, small orders or units for try-out in new machines will be forwarded at once. Design features make the Johnson especially suitable for installation in the following machinery:

Accessory Drives, Air Compressors, Bag Making, Boat Drives, Bread Wrapping, Chain Drives, Combines (farm), Conveyors, Crop Seeders, Cultivators, Dusting Machines, Feed Grinders, Floor Scrubbing, Fruit Cleaners, Gasoline Engines, Generator Drives, Hay Balers, Hoists, Loaders, Lawn Mowers, Milk Coolers, Mixing Machines, Motor Scooters, Packaging Machines, Paper Shredders, Power Fans, Power Saws, Power Take-Offs, Pumping Equipment, Sand Spreaders, Sewing Machines, Sheep Shearers, Spraying Equipment, Textile Machinery, Threshers (small), Tobacco Machinery, Tool Grinders, Tractors (garden), "V" Belt Drives, Vegetable Sorters.

Naturally, these are but a few of the possible applications for the Johnson Single Disc Clutch. The field is wide open, with new machinery constantly being developed.

Included in the driving combinations are: Gear Tooth, Bolted Plate, Pulley Type, Hub Adapter, Cut-Off Coupling Adapter, Single V-Belt Pulley Drive, and Double V-Belt Pulley Drive.

Carlyle Johnson engineers offer their engineering assistance in cooperation with your engineers and machine designers to develop the correct solution of your power transmission requirements. Write to Frank R. Simon, The Carlyle Johnson Machine Co., Manchester, Conn.

Engineering News

pressure side of the compressor. The gas is compressed from 15,000 to 100,00 psi and forced into a pressure "bomb" in which the actual experiment takes place.

The compressor consists of two sets of double-acting intensifiers driven by a 15-horsepower pump which maintains a 1000-psi hydraulic pressure. Between the low and high pressure stages, an automatic bypass system restricts action of the high pressure stage until a pressure of 15,000 psi is reached in the low-pressure stage.

•

Samuel K. Hostetter Jr. has been named sales manager of the **Crocker-Wheeler Div., Elliott Co.** at Ampere, N. J. He joined the company in 1934 and was soon assigned to the Washington, D. C. office as field engineer. For the past ten years he has served as manager of that office, supervising sales activities there and in the surrounding territory.

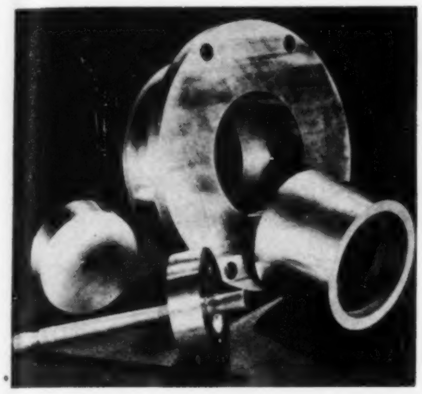
•

George S. Bond has been named sales manager of the metals and ceramics division of **P. R. Mallory & Co. Inc.**, Indianapolis. He previously served as chief engineer of the division.

•

Daniel Gurney, authority on antifriction bearings and vice president and director of engineering of **Marlin-Rockwell Corp.**, died recently in Jamestown, N. Y.

"Well, I see that engine is still shearing off the pin when it gets over 4500 rpm."



TUNGSTEN CARBIDE INSERTS: This molding die assembly used in manufacturing three-inch diameter silicon carbide and aluminum oxide grinding wheels utilizes Carboloy wear-resistant inserts. At the Carborundum Co., these tungsten carbide mold assemblies are claimed to provide an overall average working life approximately ten times that of other engineering materials. Although cost averages three times that of steel, wear-resistant qualities of the tungsten carbide metal compensate for this difference

•

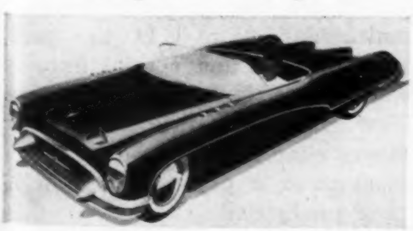
Precision Rubber Products Corp., Dayton, O., recently established a wholly owned Canadian subsidiary, Precision Rubber Products (Canada) Ltd., located at Ste. Therese, de Blainville, Quebec.

•

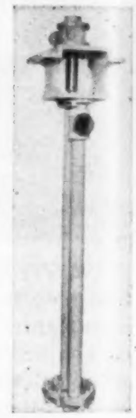
A newly incorporated firm of engineers and designers, the **Renier Co. Inc.,** has established offices at 5209 Euclid Ave., Cleveland. The firm will undertake the complete design of machine tools, hydraulic presses and equipment, and specialized machinery of all types.

•

PLASTIC BODY: To test the possibility of using glass fiber laminates in automobile bodies, a new sports convertible has been introduced by Buick Motor Division. The lightweight, single seat experimental Wildcat is mounted on a 114-inch wheelbase and is powered by a 188-horsepower V-8 engine



Rotary Air Motor Withstands High Temperature



Used once in a test location so hot that lubrication was supplied from a lubricator 20 feet away to prevent the oil from boiling, a 3-horsepower Gast rotary vane air motor drives a Wemco metal pump completely immersed in molten metal in the unusual application shown. The pump emptied a furnace containing 8400 pounds of aluminum at 1360 to 1400 F.

Limited mold capacity required starting and stopping the pump several times. Actual pumping time was less than four minutes, although the pump remained in the molten metal for six hours.

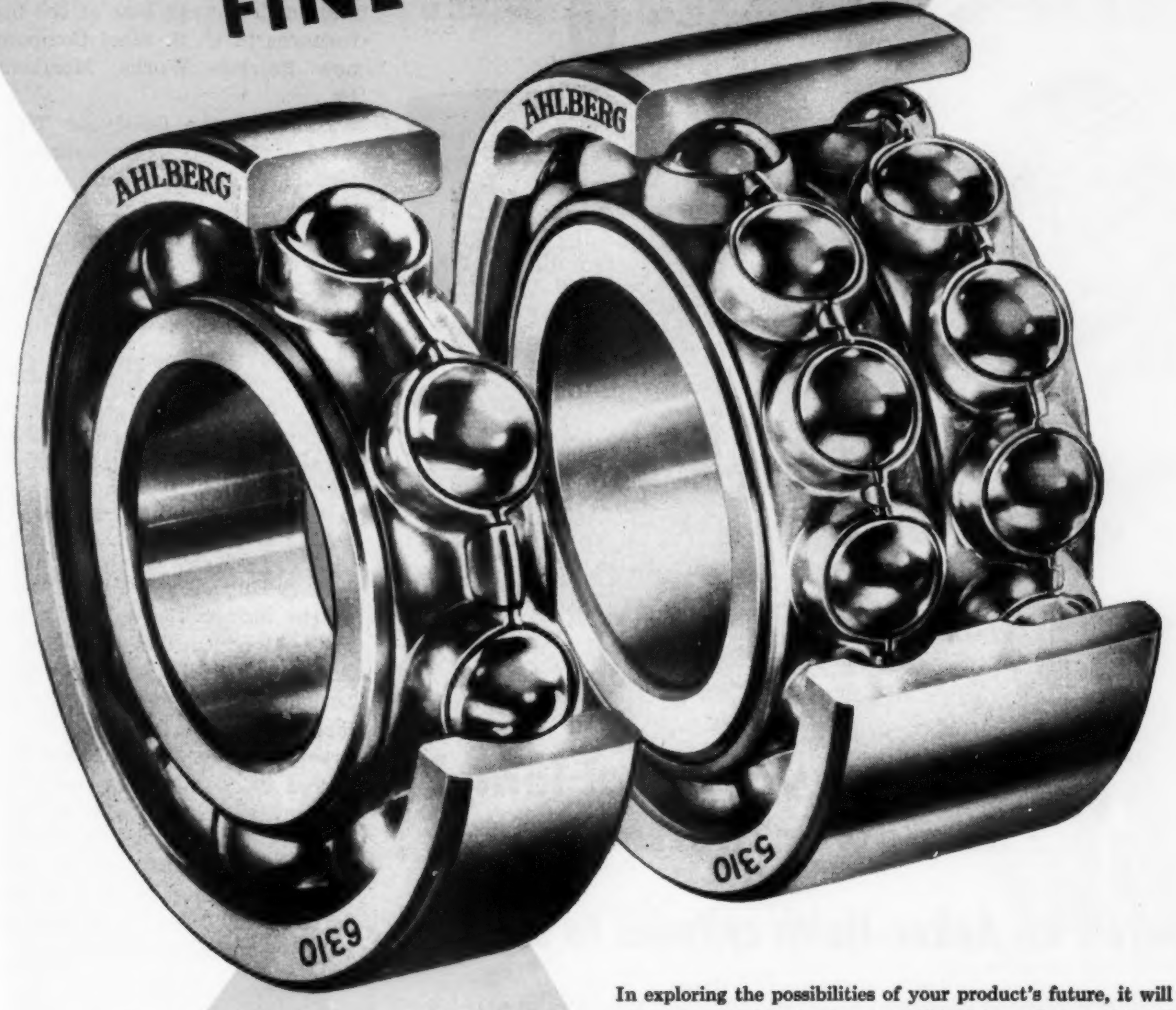
Since the furnace could not supply its own heat after it had been partially drained, an oil burner was directed on the surface of the liquid. The burner flame, directed on the pump body, caused the gray iron casting supporting the motor to reach cherry red heat.

An inspection after the test revealed that the motor vanes were unharmed, although some carbonization of the lubricant in the bearings was present.

♦

Announced recently is the formation of a private company to design and manufacture noncircular gears, founded by **Fred W. Cunningham**, consultant to the Arma Corp. Basis for the business is a standard gear shaper operated by a motion picture film which has been automated by Dr. Cunningham. Engineering and manufacturing services will be provided in connection with noncircular gears. In the past, steel masters and special fixtures were needed to cut such gears. Now fast, accurate, automatic means eliminate the need for metal masters as a preliminary step in their production.

Fine machines begin with FINE BEARINGS



In exploring the possibilities of your product's future, it will pay you to remember this long established fact: Fine machines begin with fine bearings—and fine bearings mean Ahlberg Bearings.

For over 45 years design engineers have found that Ahlberg Bearings offer greater opportunities for improving design, stretching equipment life, reducing maintenance and operating costs.

Ahlberg Ball Bearings are available in both self-contained and mounted units—in all conventional sizes. Write for complete data or engineering consultation on a no-obligation basis. Ahlberg Bearing Company, 3025 W. 47th Street, Chicago 32, Illinois.

AHLBERG

BALL BEARINGS

ENGINEERED AND BUILT FOR THE WORLD'S FINEST EQUIPMENT

AUTOMOTIVE . . . INDUSTRIAL . . . AIRCRAFT . . . MARINE . . . ARMAMENT

VISIT US AT THE INTERNATIONAL PETROLEUM EXPOSITION—TULSA—BOOTH 12, CALIFORNIA BUILDING

Engineering News

Longest Conveyor Belt Delivers Coke to Furnaces

The world's longest single-flight, rubber and fabric conveyor belt hauls quenched, sized and screened coke from the coke plant screening station to storage bins at the blast furnaces in U. S. Steel Company's new Fairless Works, Morrisville, Pa.

Built by the Goodyear Tire & Rubber Co., the belt measures 5500 feet from the center of the head pulley to the center of the tail pulley. The 11,000 feet of belting used in the conveyor is 4 feet wide and is constructed of especially compounded synthetic rubber and rayon fabric.

Coke carried on the belt weighs 30 pounds per foot and registers temperatures of approximately 250 F as it is being moved 300 feet per minute at a capacity rate of 400 tons per hour to the bins.

◆

Sam Dupree, assistant manager of the industrial products division of the **Goodyear Tire & Rubber Co.**, Akron, since 1947, has been named assistant to R. S. Wilson, vice president in charge of sales. Mr. Dupree joined Goodyear in 1934, transferred to mechanical goods development, and in 1939 was named development manager at the new molded goods plant at St. Marys, O. He became sales manager there in 1945, and held that post until returning to Akron in 1947.

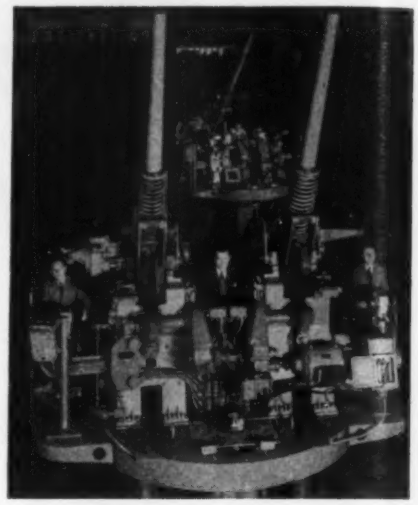
◆

Theodore V. Purvin has been appointed general sales manager of **Amgears Inc.**, Chicago.

◆

Carl E. Schmitz was recently appointed vice president of sales for **Crane Packing Co.**, Chicago. He was formerly vice president and director of engineering. A reorganized sales department under the direction of Mr. Schmitz will include five division sales managers: E. H. Stubenrauch, mechanical packings; Stephen Hawxhurst, molded Teflon products; Harry I. Sole, Lapmaster; V. E. Vorhees, mechanical seals; and Stillman Segar, plastic lead seal.

Engineering News



GUN MOUNT: Carrying a pair of three-inch antiaircraft guns, this electronically controlled gun mount is the first produced by the Firestone Tire and Rubber Co. under a $62 million U. S. Navy contract. Weighing 17 tons, the mount is equipped with automatic loading devices and radar fire-control systems. During World War II, Firestone manufactured more than 30 thousand 40-millimeter gun mounts for the Army and Navy, as well as 90-millimeter cannon

•

The **Resistance Welder Manufacturers Association,** in its annual Prize Paper Contest, is offering a total of $2250 in prizes for outstanding papers dealing with resistance welding subjects. Papers may be submitted in three categories: industrial class, university staff class or university undergraduate class. The contest is open to anyone in these categories living in the United States, its possessions and Canada. Judges will be appointed by the American Welding Society, and awards will be made at the fall meeting of the society. Entries will be accepted until July 31. Complete details regarding subject matter of the papers and other contest rules may be obtained from the Resistance Welder Manufacturers Association, 1900 Arch St., Philadelphia 3, Pa.

•

At the annual meeting of **The Aluminum Association,** D. A. Rhoades, Kaiser Aluminum & Chemical Corp., Oakland, Calif., was elected president for this year.

Stress Relief

AGAIN J. P. Henderson returns with a controversial issue. He concludes, you will see, with a question that might provoke argument.

The Huddle Engineer

An organization should be a co-operative enterprise. This is trite enough so that you are welcome to your "ho-hum" at this point. But perhaps there are some aspects to this co-operation that you had not considered before.

Take, for instance, the relationship between the department head and his employees. On certain points they are on opposite sides of the fence. You want a higher salary; he is under pressure to get the best engineering he can for the money. In fact, his executive effectiveness is measured partly in this very manner. Here psychologically, are all the elements of clash. But in certain other fields you and the boss are on the same side, or should be. It's a matter of presenting a united front to the rest of the organization, and to the outside world. And there is where the co-operation comes in.

What is expected of each in this co-operative deal? Well here are some of the things the boss should do for you.

He should see to it that your increased efforts or improved work are rewarded without too much delay. (This may require more battling with the management than you think.)

He should protect your salary position by not hiring new men of less experience at more money than he pays you.

In meeting with the organization or with outside personnel he should cover up your mistakes as best he can, putting as good a face as possible on the grief you might have caused. The scolding you deserved should be reserved for a later private conference between the two of you.

He should see to it that work

is divided with reasonable equity. It is disconcerting to belong to an organization in which certain persons are largely idle and others are overloaded (especially if you are the one overloaded).

He should provide you with all of the tools and conveniences comparable with the rest of the organization. (I once knew an organization in which the engineering and the sales offices were located adjacent. In sales the offices shined with expensive furniture. The engineering department was furnished with broken-down tables and battered desks. The chief engineer had never had the courage to demand a better deal.)

There's the list of what the boss can do for you.

What can you do for the boss?

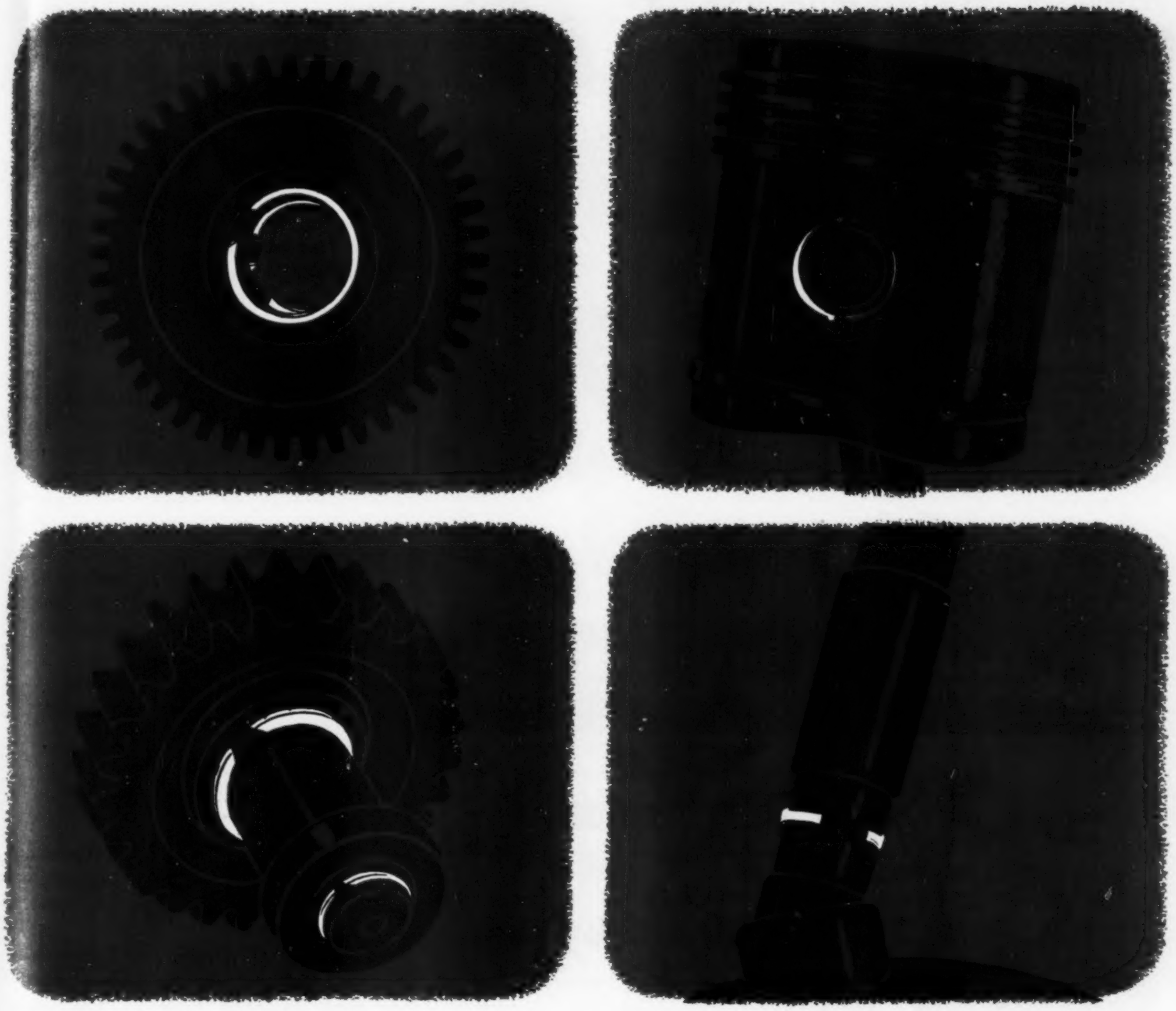
This month, I want to illustrate just one item by describing the huddle engineer—a species you may recognize.

This is the man who never played on the football team, but observed the huddle system and brought it into the office. Periodically he gathers some of his pals to hear the latest story, making a conspicuous group. If some of his pals are ex-cheer leaders and some of the office girls are equipped with high resounding giggles, the combination is ideal for disrupting work and getting the boss in bad.

To a greater or lesser extent, every office seems to come equipped with some of this type. The boss has two choices. He can break it up. He thereby becomes a hard-boiled slave driver and that spirit of give-and-take essential to smooth personnel relations can be lost. Or, he can overlook it at the risk of his own reputation, absorbing the scoldings of his own boss and the sly looks of other officials.

It should not be a surprise to employees that their own boss is not an absolute ruler. Yet they apparently lack the imagination to realize that he depends upon a reputation for effectiveness just as much as they do. The huddle system in operation when another department head, the company president, or a member of the board of directors walks through the department, can result in his being called on the carpet. Or, if that

Cut Your Machining costs...



with Reliance RINGS

As an engineer or designer, you gain sizeable manufacturing economies by specifying Reliance RINGS for your shaft assembly, bearing applications or counterbore designs. By machining grooves in the shaft (instead of machining the entire shaft) and applying Reliance Rings to form shoulders for locking bearings, gears or other parts into proper postion—you immediately save material, machine time and labor costs! In like manner, you eliminate counterboring to form internal shoulders. To meet every possible application, Reliance Rings are available in carbon, alloy, stainless steel, aluminum and non-ferrous metals to specified physicals, cross sections and diameters. Gain the savings hundreds of manufacturers have realized by designing Reliance Rings into your product—Today!

Have the complete facts available on Reliance Rings in your department—call your nearest Eaton Reliance Sales Office or write direct for Engineering Bulletin 4k/3.

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Stress Relief

doesn't work at once, the next time the boss asks a favor—more money for the department, more equipment or what-not—he can be told very coldly that he is not using his present personnel effectively. Just about then, the boss is ready to strangle with bare hands the source of that next loud guffaw and far-carrying giggle.

If the boss is not completely a stiff shirt, he usually views the members of the huddle more calmly, but with little respect for their intelligence. They lack the imagination to see the position they place him in. Why can't they arrange to do all this less obviously?

No realistic employer expects continuous work from his staff. The best he can hope for is that they work the biggest part of the day, co-operate on rush jobs when the pressure is on, but nearly always manage to look busy.

Is this personal dishonesty?

—J. P. Henderson

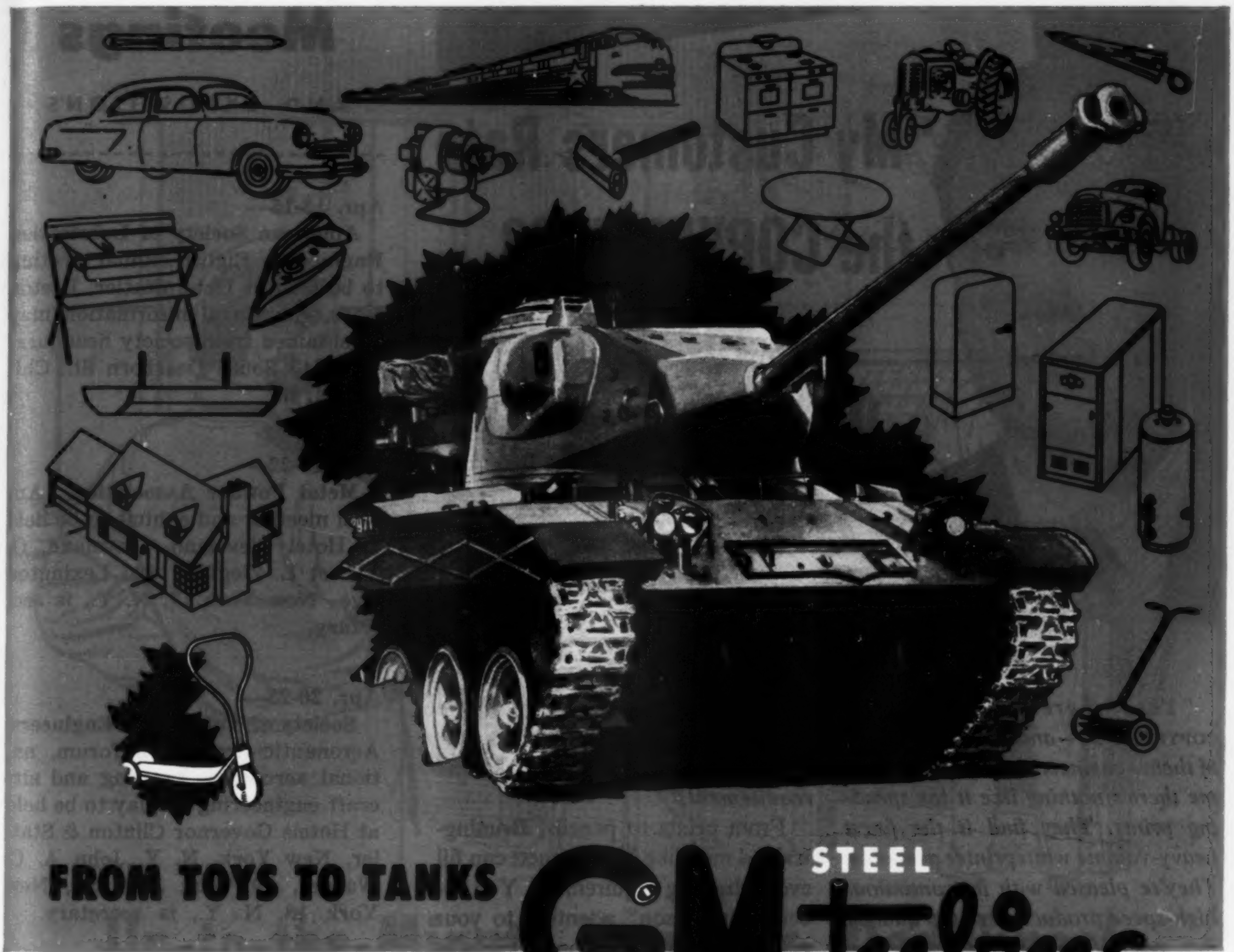
They say . . .

"We must look to engineering superiority for our defense because in manpower we are hopelessly outnumbered.

"The U. S. Bureau of Labor Statistics data indicates that the current national pool of engineers and scientific personnel is approximately 400,000. Of these, 10,000 per year retire, die, or fade away; 15,000 per year are required by the armed services. We estimate that 30,000 new graduate engineers are needed each year, exclusive of the military.

"We can expect only about one-half this number. Based on the present enrollment in engineering colleges, the U. S. Bureau of Labor indicates that there will be approximately 22,000 graduates in 1952 and 17,000 to 18,000 in both 1953 and 1954.

"If this happens, the United States stands to lose its technological supremacy." — Richard O'Mara, *Representative of Engineering Manpower Commission, Western Precipitation Corp., Los Angeles, Calif.*



STEEL GM tubing

SOLVES PRICE AND PRODUCTION PROBLEMS

WIDE RANGE OF SIZES AND FINISHES

GM Steel Tubing is available in welded single wall or copper brazed double wall construction, in sizes from 1/8" to 3/8" O.D., and in straight lengths or in 120' to 1000' continuous coils. Choice of plain, Terne coated, copper plated, or Fuse-Kote (copper fused) finish.

EASILY FABRICATED

GM Steel Tubing not only lends itself readily to intricate bending but also to all types of fabricating operations—crimping, beading, perforating, swaging, expanding, drawing, flanging, flattening, knurling, threading. Rochester Products offers complete fabricating facilities, including the services of experienced General Motors engineers.

It's a far cry from a child's toy to a mighty weapon of war—yet GM Steel Tubing plays an important part in the design and building of both. In hundreds of other products, too, this low cost steel tubing serves to advantage either for the passage of liquids or gases, or as part of the mechanical structure.

Every day Rochester Products turns out ***hundreds of miles*** of GM Tubing in one of America's most modern and best equipped plants. Into every inch of its manufacture go the skill and engineering "know how" gained through nearly 20 years of General Motors research and development.

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Meetings

AND EXPOSITIONS

Apr. 13-15—
American Society of Lubrication Engineers. Eighth annual meeting to be held at Hotel Statler, Boston Mass. Additional information may be obtained from society headquarters, 343 South Dearborn St., Chicago 4, Ill.

Apr. 20-22—
Metal Powder Association. Annual meeting and exhibit to be held at Hotel Cleveland, Cleveland, O. Robert L. Ziegfeld, 420 Lexington Ave., New York 17, N. Y., is secretary.

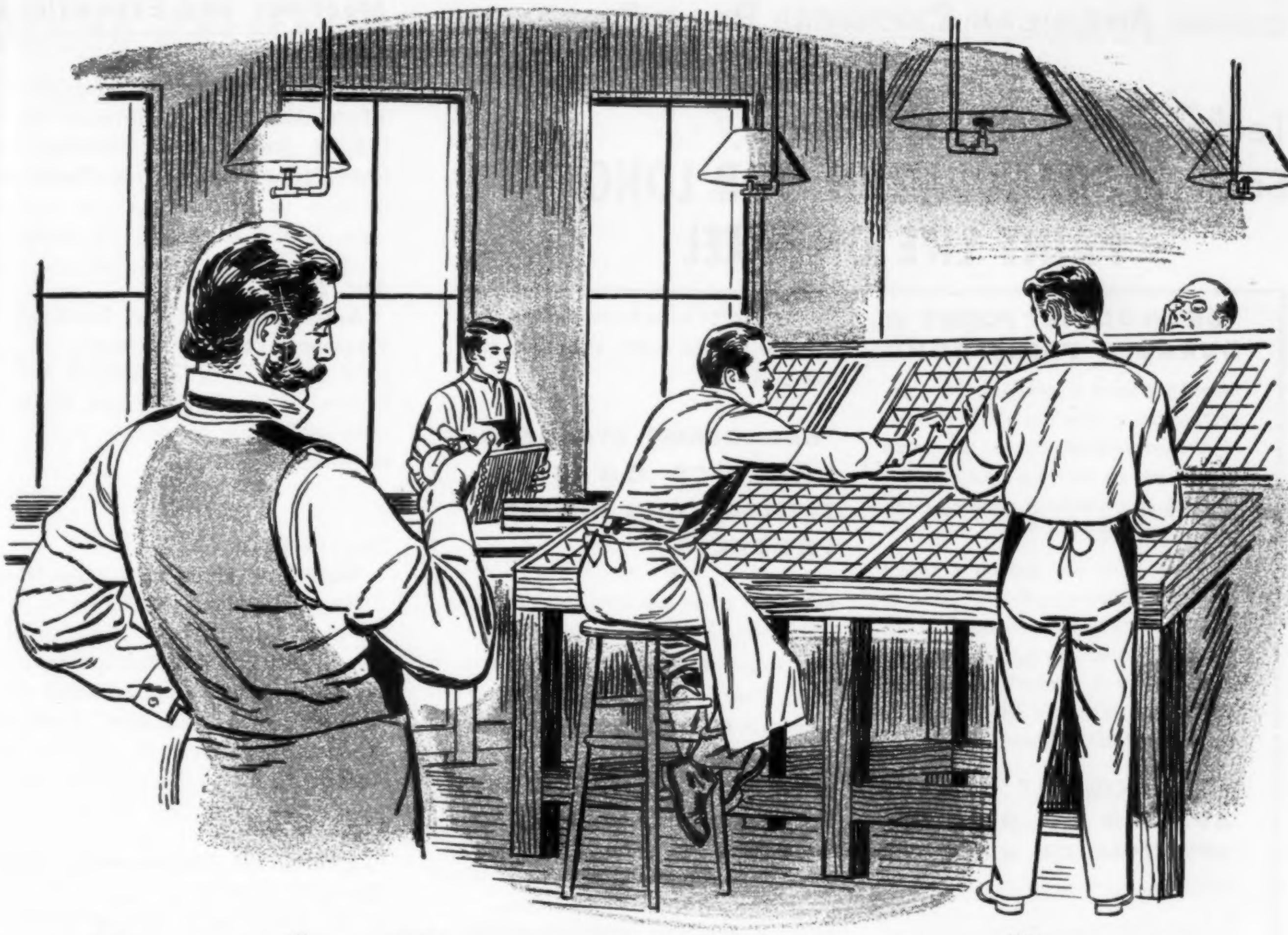
Apr. 20-23—
Society of Automotive Engineers. Aeronautic production forum, national aeronautic meeting and aircraft engineering display to be held at Hotels Governor Clinton & Statler, New York, N. Y. John A. C. Warner, 29 West 39th St., New York 18, N. Y., is secretary.

Apr. 20-23—
Packaging Machinery Manufacturers Institute. Annual packaging exposition to be held at the Navy Pier, Chicago, Ill. Additional information may be obtained from society headquarters, 342 Madison Ave., New York 17, N. Y.

Apr. 26-30—
American Ceramic Society. Annual meeting to be held at Hotel Statler, New York, N. Y. Charles S. Pearce, 2525 North High St., Columbus 2, O., is secretay.

Apr. 27-28—
American Zinc Institute Inc. Thirty-fifth annual meeting to be held at Hotel Statler, St. Louis, Mo. Ernest V. Gent, 60 East 42nd St., New York 17, N. Y., is secretary.

Apr. 27-May 8—
British Industries Fair to be held at Olympia and Earls Court in London and Castle Bromwich,



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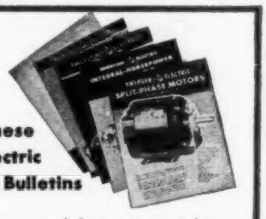

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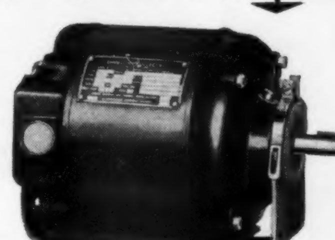
Manufacturers requiring motors 1/20 to 5 h.p. can profitably use these reference guides. Specifications, construction and performance data are included for these motors:

- ☐ M401-A Capacitor-Start
- ☐ M401-B Split-Phase
- ☐ M401-C Integral
- ☐ M401-D Fan-Duty
- ☐ M401-E Oil-Burner
- ☐ M401-F Jet Pump
- ☐ M401-G Blower



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Birmingham, England. Additional information may be obtained from British Information Services, 30 Rockefeller Plaza, New York 20, N. Y.

Apr. 28-30—
American Society of Mechanical Engineers. Spring meeting to be held at the Deshler-Wallick Hotel, Columbus, O. C. E. Davies, 29 West 39th St., New York 18, N. Y., is secretary.

May 18-22—
Materials Handling Exposition. Fifth national exposition to be held at Convention Hall, Philadelphia, Pa. Additional information may be obtained from Clapp & Poliak Inc., 341 Madison Ave., New York 17, N. Y.

May 21-23—
Society for Experimental Stress Analysis. Spring meeting to be held at the Hotel Schroeder, Milwaukee, Wis. W. M. Murray, Box 168, Cambridge 39, Mass., is secretary-treasurer.

May 24-28—
Scientific Apparatus Makers Association. Annual meeting to be held at the Greenbrier Hotel, White Sulphur Springs, W. Va. Additional information may be obtained from society headquarters, 20 North Wacker Drive, Chicago 6, Ill.

May 27-29—
American Society for Quality Control. Seventh annual convention to be held at Convention Hall, Philadelphia, Pa. Additional information may be obtained from society headquarters, 70 East 45th St., New York 17, N. Y.

June 15-19—
Basic Materials for Industry Exposition. First exposition and conference to be held at Grand Central Palace, New York, N. Y. Additional information may be obtained from Clapp & Poliak Inc., 341 Madison Ave., New York 17, N. Y.

Design Abstracts

(*Continued from Page* 273)

in *Fig.* 3. Equations based on the fundamentals of hydrodynamics permit the evaluation of supply pressure, P_0, volume of flow Q, and friction torque, T. These equations are

$$W = \frac{P_0 \pi}{2}\left[\frac{R^2 - R_0^2}{\ln\left(\frac{R}{R_0}\right)}\right] \quad (3)$$

$$Q = \frac{P_0 \pi h_0^3}{6\mu \ln\left(\frac{R}{R_0}\right)} \quad (4)$$

$$T = \frac{2\pi\mu\omega}{h_0}\left[\frac{R^4 - R_0^4}{4}\right] \quad (5)$$

An interesting application to an ultracentrifuge is shown in *Fig.* 4. The vertical load is supported on an air step bearing and the rotor is spun by air jets. This unit operates at a speed of approximately 80,000 rpm. The friction is so low that once the case is evacuated, and the driving air supply shut off, the rotor will continue to spin for many hours before coming to rest.

Hydrostatic lubrication itself generally leads to very low coefficients of friction, even with oil. A rather striking example is that of the Hale telescope on Mount Palomar, where the overall coefficient of friction, using a medium-viscosity oil, is 0.000004. This telescope, weighing about one million pounds, is rotated by a 1/12-hp clock motor which actually supplies more power than is required.

A recent design for an electrical dynamometer with a cradle weighing 1300 pounds, reports a coefficient of friction of 0.00000075. Actually this is zero for all practical purposes—and was obtained with oil. Air under similar conditions would introduce even less friction.

Compressible Flow: It should be emphasized that the equations set forth here are based upon the hydrodynamic theory of incompressible liquids and can only be used with air if the pressures involved

(*Continued on Page* 384)


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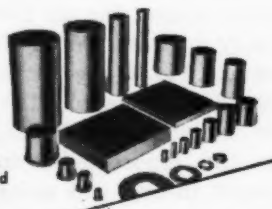

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Design Abstracts

(*Continued from Page* 381)

are not greater than a few pounds per square inch. Where greater loads are carried and unit pressures are higher, the analysis must include the compressibility effect due to significant changes in volume as the air passes through the bearing.

As an example of this type of calculation, consider the step bearing again, ***Fig.*** **3**. With air as the lubricant and assuming isothermal expansion for the flow in the floor of the bearing,

$$V_0=\frac{\pi h_0^3}{6\mu\ln\left(\frac{R}{R_0}\right)}\left[\frac{P_0^2-P_1^2}{2P_0}\right] \qquad (6)$$

where P_0 and P_1 are absolute pressures and the volume V_0 corresponds to inlet conditions (P_0). Equation 6 for compressible isothermal flow should be compared to the equivalent Equation 4 for incompressible flow.

Bearing Developments: Some further considerations regarding air-lubricated bearings have been found in a recent German paper. The conically-shaped bearing supported by air under hydrostatic pressure shown in ***Fig.*** **5** is taken from the paper. Air is supplied through the circumferential groove at the middle of the bearing or through a belt of holes at the same location. This is in reality an elaboration of the step bearing described earlier. Before, with the simpler design, the air flowed only radially outward. With the design shown in ***Fig.*** **5** the air flows both outward and inward. The authors have tested this and similar bearings and report good correlation between theory and measurement.

It should be obvious that air-lubricated bearings can be constructed in a number of forms to match the peculiar operating demands of a wide variety of applications. Air bearings have certain major advantages for many types of work. Within their limitations then they do represent a relatively new solution to the general problem of bearing design and analysis. Our experience has already indicated that the range of application of air lubrication is

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From a paper entitled "Low Friction Properties of Air Lubricated Bearings" presented before the New York Academy of Sciences in New York, N. Y., January 1953.

Feedback Control Opens New Horizons

By Gordon S. Brown

Head, Department of Electrical Engineering Massachusetts Institute of Technology Cambridge, Mass.

FEEDBACK control is not something brand new. On the contrary, it is as old as engineering itself. Lord Kelvin wrote about it, and James Watt applied it to the flyball governing of his steam engines. Its early industrial applications, however, were usually manual or semiautomatic. A human operator read instruments—a temperature or pressure gage, for example—and applied a correction to a process by turning a valve or adjusting a rheostat. He served as the feedback link, the error detector and the controller. But throughout the years the consistent increase in the tempo of industry brought about greater and greater complexity of equipment, greater need for precision and faster response of controllers.

Eventually the human was ruled out of the control loop by the addition of amplification to the instruments to let them replace the action of the human muscle. Instruments which formerly had merely measured and given data for inventory purposes now measured, indicated, and controlled. Complex decisions and judgment that had been made by the human brain were soon performed, rather imperfectly perhaps, by mechanisms designed to approximate certain of the elementary human actions such as proportioning, anticipating and integrating. These steps were very significant throughout all technology for they freed the machine from the limitations imposed upon its operations by the shortcomings of the physical and nervous systems of

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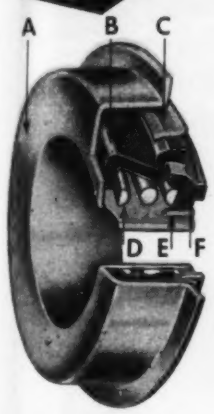
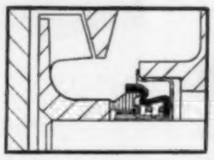




A. Seal Retainer.
B. Synthetic Rubber Bellows.
C. Metal Band.
D. Spring.
E. Holding Dents.
F. Precision-Lapped Sealing Washer.

STATIONARY SEALING HEAD TYPE 11-A

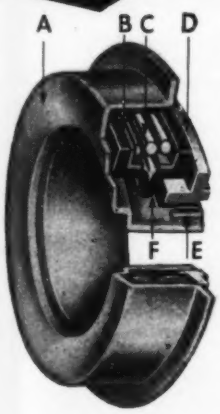
This pressed-in packaged sealing unit is designed to protect the spiral steel spring inside of the synthetic rubber bellows against corrosion.

The spring clamps the flange of the bellows tight against the bottom of retainer and the metal ferrule squeezes the top portion of the bellows against the "Teeplelite" washer.

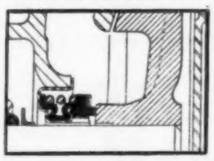
This liquid-tight seal is especially recommended for small shafts on *hot or cold water, oil, gasoline, kerosene, soapy water.*

Use this seal on: *Low Pressures* 35 PSI
Low Temperatures −65°F. to 220°F.

LOW COST



STATIONARY SEALING HEAD TYPE 6-A



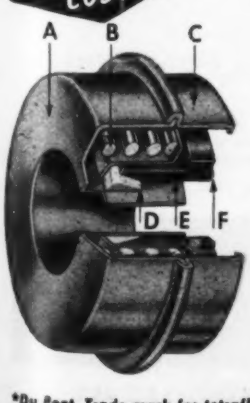
A. Seal Retainer.
B. Metal Ferrules.
C. Spring.
D. Precision-Lapped Sealing Washer.
E. Holding Dents.
F. Synthetic Rubber Bellows.

This pressed-in packaged sealing unit is designed with a spiral steel cadmium plated spring (stainless or bronze optional) assembled between two metal ferrules which clamp the flanges of the synthetic rubber bellows tightly against the retainer shell and the "Teeplelite" washer.

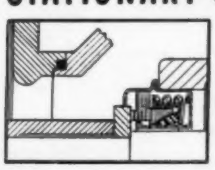
This liquid-tight seal is especially recommended for small shafts on *hot or cold water, oil, gasoline, kerosene, soapy water.*

Use this seal on: *Medium Pressures* 75 PSI
Low Temperatures −65°F. to 220°F.

LOW COST



STATIONARY SEALING HEAD TYPE 9-A



A. Seal Retainer.
B. Spring.
C. Sleeve.
D. Wedge of Teflon.
E. Sealing Washer.
F. Precision-Lapped Face.

This pressed-in packaged sealing unit is designed to use a wedge sealing ring of Teflon*. This wedge shaped sealing ring closely fits the inner sleeve of the retainer and makes a liquid-tight contact with cone surface of carbon washer.

The result is a liquid-tight seal which is especially recommended for *hot oil, and all chemical liquids or gases, hot or cold.*

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*Du Pont Trade-mark for tetrafluoroethylene resin

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the human. Thus was set loose a tremendously powerful technique that has resulted in the birth of feedback control. Throughout all industry, processes are daily being re-examined to see how automatization by feedback control systems can help keep pace with our expanding technology.

Present State: The key to the modern state of control is actually not distinguishable unless we look behind the maze of motors, pistons, wires, electron tubes and brightly lit panels of instruments. What we find is a philosophy, a frame of mind, an engineering methodology of how to design and synthesize control systems.

Feedback system engineering, as considered here, is the co-ordinated creative synthesis of the process, the plant and the instruments. Developments predicted for the future call for the creative co-ordination of teams of physicists, mathematicians, and technical specialists of many kinds, and culminate in engineering in which system synthesis over-shadows analysis. Feedback control system engineering, as practiced under the demands of the mid-twentieth century, deals almost entirely with techniques that are new. It is not routine engineering. It is often a highly technical administrative job. It offers the highest of professional challenge.

The groundwork for the feedback control system engineering techniques that exist today was laid at about the time the flyball governor was invented, but no particular merger of theory and practice took place until about the 1920's. Then came a gradual emergence of technological method. First, men began to formulate and to solve mathematical equations that designated the behavior of the control systems they intended to build. Often the results were rather surprising. As the systems became more complex, man's capacity to guess correctly did not bear mathematical scrutiny. Slowly from the mid-twenties until the late thirties, the methodology grew and influenced control system designs, but even when World War

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II began, we were barely muddling through. Statics had not yet given way properly to dynamics in our thinking.

As the general theory for the analysis and synthesis of control systems emerged, it became apparent that the theory itself was not enough. Some basis had to be found for speeding up the solution to design studies. The old era of the cut-and-try gadgeteer passed—the predetermined and predicted gadget became more frequent. A century ago many problems could be solved with relatively simple controllers which might be constructed by a clever inventor or technician. World War II and subsequent controllers, however, demanded more exacting knowledge of the device or process to be controlled and of the basis for setting reasonable specifications. The coupling between science and engineering now became very tight as the designer probed more deeply into the basic theory of processes. The calculations were tedious and long so charts, nomograms and sliderules were invented for the solutions of special problems. Criteria of dynamic excellence began to emerge.

Research Stimulated

But a great vacuum still existed in the numerical data available to the engineer. Its presence stimulated whole classes of experimental testing studies on live systems — mechanical, electrical, chemical and aeronautical. Complete plants or mechanisms of systems were disturbed by test signals, and the response compared with the disturbance. It became convenient to write the describing equations for electric motors, amplifiers, hydraulic drive and even chemical processes in terms of their message-handling ability and to analog them by means of passive and active circuits containing resistors, capacitors, vacuum tubes and other simulation apparatus. By the application of the transient and sinusoidal test function, the use of operational mathematics, the exploitation of signal-flow diagrams and block

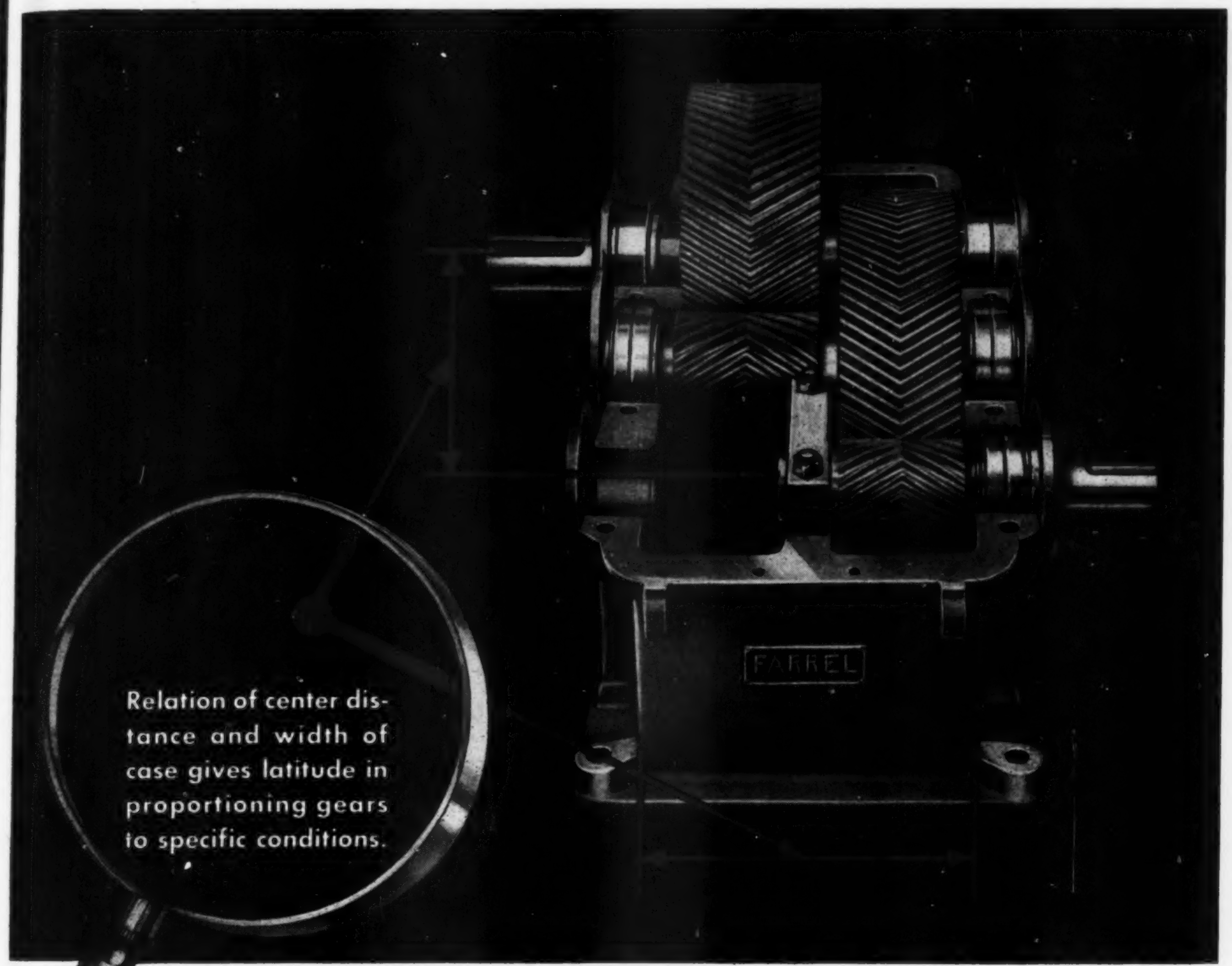
Design Abstracts

diagrams, and the complex variable forms for transfer functions, the system engineer brought the whole vast and important theory of communications and electrical circuit analysis to bear upon the analysis and synthesis of feedback control.

The changes in equipment design that today are upsetting the peace of the industrial and military scene stem from the energetic collaboration of bold and imaginative engineer-scientist teams. Today's new apparatus is gone tomorrow. Improved performance of a control system and great new systems of control no longer come into being merely by men at drafting boards plying themselves diligently to variations upon components. Rather they begin with the system engineer who draws up a unified scheme for control whether his subject be a missile or an industrial process. He pulls together his team of abstract thinkers, scientists and creative engineers. He poses such questions as: "Are we exploiting magnetic phenomena so that the dynamics of our energy conversion members respond in an optimum manner to the most data we can send over our channels, or is it possible that by matching process-reaction dynamics with instrument dynamics we can have less hold-up in our process and hence not build such massive structures?"

Gadgeteering Outmoded

The rise of ideas which have led to the frame of mind, the art so to speak, developed because system engineers found a need to be more quantitative as they were faced with the tasks of processing more and more material in less time with less space, and to closer tolerances. Their practical problems had become more complex, more sophisticated. They were face to face with the expanded tempo of industry. They wanted numbers to measure physical phenomena, they wanted to replace the guesswork and cut-and-try gadgeteering by procedures based upon modern scientific discoveries, and mathematical law expressive



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of the dynamic state of system behavior. They had to substitute exact design with confidence for gadgeteering and empiricism. They became vitally interested in the useful aspects of modern science. They learned that analysis must precede construction, especially for systems so complex that they take five or ten years to build and thus often outspan the engineering and managing ability of any one group of men involved in the task. They know that the creation of the systems demanded by the future will embrace even greater scope and complexity and call for the effort of closely coupled teams of widely different specializations whose knowledge of the modern discoveries of science, engineering, and management is first-rate.

Today's Outlook: On all fronts automatic control is flourishing, so much so that we can be pardoned if we view the race with alarm. The robots are here. Our schools are turning out the engineers to design their components and to conceive of them in toto. Industries are springing up every month to build heretofore unheard of mechanisms. Older industries have shifted over to new lines of equipment. Researchers are probing new frontiers. Conferences almost every few months are devoted exclusively to the technical and scientific aspects of the problem.

Now, unlike a century ago, theory and practice seem to be advancing to some extent in synchronism, though the deeper we probe the more we realize that conventional methods of analysis are inadequate. Thus, the pressure of physics and mathematics still continues. The physics laboratory has moved into the production plant. Today it is not enough to know the temperature and the pressure under which a chemical reaction may be taking place. We seek to know the molecular weight of the material being made. We search for measures of its color, its odor, its ability to transmit light, its infrared absorption, the amount of moisture taken up by the material. So men have gone to higher strata of measurement.

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C-D-F has mass production facilities for both the manufacture and fabrication of eighteen distinct grades of Spiral Tubing. For example, there's a special punching grade, fine for punching rectangular or square holes near the end of the tube. A relatively soft tube is supplied for difficult stapling or riveting. C-D-F makes high strength automotive electrical bushings from a very hard tubing with high axial compressive strength. Combinations of kraft, chipboard, Diamond "fish paper" Insulation, and other materials are available.

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They shoot missiles into the sky, they look at materials with special radiations, they spin propellers in the streams, they beam sound-waves around the vats and pipes, all in an effort to become more quantitative about the dynamics of what they are doing. All this calls for great amounts of study about what to do with the measurements when the data are available. After all, it is one thing to decide how to measure the viscosity of a fluid; it is another thing to know what knobs on the process to turn on or what ingredients to add to change the stickiness of the fluid an exact amount, yet these are the frontiers being attacked today.

Training Required: Feedback control system engineering is now here as a new rapidly growing profession. It is one that demands a type of system thinking that ignores traditional categories in both educational and industrial organizations. The training of young men for this new level of professional competence requires a reorientation of our ideas of post-graduate engineering education. The proper utilization of this new kind of talent requires an awakening and a reorientation of the attitudes of engineering management. At all management levels—financial, technical, operating and maintenance—we must be prepared to recognize control with its benefits and its limitations. Today we have a great paradox. We almost cannot afford not to have lots of control. However, compromises must be made between costs and performances. Old methods of design are being forced to yield to new ones with closer coupling between instrumentation limits and the costs and strength of steel tanks. Finally, men who have been used to operating the controllers of 1930 vintage are receiving more and more often black boxes of the 1950 vintage. They now must learn new tricks of why they work, how they work, and how they can be repaired. Thus, the student of industrial management, the engineer involved in research, development and design,

Design Abstracts

and the man who keeps the equipment running, all must become conversant in a complementary way with feedback control system engineering.

Feedback System Engineering: Control system engineering requires first, an analysis of the whole system in terms of independent, partially independent and dependent variables; second, an integrated design of instrument, process reaction and controller; third, which is too often the first, the initiation of mechanical design, fabrication; and finally, the test. Perhaps the airplane is the unique example to cite here. The modern jet fighter is no longer merely an exquisite airframe but a unique optimization of a weapon, a killer, and a navigable flying machine. The coupling required between all branches of science and engineering for its creation is extremely close.

The concepts of automatic control are now developing from those that call for the mere assembly of standard components, to the synthesis of unique components evolving along the pattern of servomechanisms design. Such further steps as the incorporation of computational aids in automatic control for the simultaneous control of several variables from combinations of environmental and quality information, in such a way as to require the instant-by-instant solution of complex groups of equations during the operation of a plant or system are clearly indicated trends. In some cases, instead of a steady-state condition of operation it is found better to use a continuously adjusted or sliding condition. Such requirements often call for computers as part of the system.

Particular stress must be laid on the science and practice of measurement, which is perhaps the most important topic in the automatic-control field, first, in order to ascertain the proper quantity to be measured and second, to prescribe the dynamic accuracy required. The measurement of product quality represents chal-

lenging new areas of basic research, extending to the field of nuclear techniques. The field of electronics and electromagnetics appears to offer more opportunities for improved measurement and amplification of signals than any other means. Vacuum-tube amplifiers, pulse techniques, storage and memory techniques are in everyday use and must be included with a student's training.

Familiarity with mathematics of differential equations, functions of a complex variable, statistics, and non-linear techniques, as well as knowledge of physics and chemistry of the mid-twentieth century variety are basic. A vital, but much neglected field, is that of energy conversion and energy transfer, in relation to dynamic processes. Computational aids such as differential analyzers, computers or simulators, are needed in the analysis and synthesis of full-scale problems, though they are no substitute for keen analytical thinking. Contact with these devices are vital parts of training.

Tomorrow's Outlook: Even in the most robotized of the automatic factories there will be many men. We are far from the time when men can be replaced. Nor should labor fear the widespread growth of control. On the other hand the use of control coaxes him none too gently into taking more responsible jobs, making bigger decisions and using his mind as well as his hands. On the other hand it gives him greater freedom and leisure for hitherto unreachable creative pursuits.

To many thinkers, feedback as an evolving concept is as important as anything now in sight. If the atmosphere in which our study of machine systems takes place is profound and invigorating, we should be able to give ourselves the collatoral capacity to think about problems arising in human systems. Already feedback as a concept has given us a new insight into the nature of man and the operations of his body. Over and above these virtues we find the need for engineers to apply this kind of imaginative think-

ing to their actions—both long and short term—as citizens. Engineers as a class should always think comprehensively about the effect on society of their own actions if engineering is truly a profession.

It should not be out of place here to contemplate human society as a large feedback system wherein man's intellect is the feedback link between his environment and his conscience. If true, it follows that the attitudes of mind we are trying to encourage in the various technical fields of feedback control may be the key to a new type of thinking and a new way of looking at the social problems of the twentieth century.

From a paper entitled "Feedback System Engineering—An Expanding Field" presented at the AIEE Centennial of Engineering Meeting in Chicago, Ill., September 1952.

A Look at Gear Scoring

By B. W. Kelley
Research Engineer
Caterpillar Tractor Co.
Peoria, Ill.

UP TO recent years, occasional gear scoring problems were subdued without much technical difficulty, although always with costly production changes, by modifying gear geometry in a trial and error manner or changing the oil characteristics.

A scored gear will not put a machine immediately out of operation but the damage caused by the apparently superficial failure is threefold. First, it produces a noisy pair of gears. Second, small particles of metal released attack oil systems and aggravate the wear of bearings and other gears in the same oil supply. Third, gear tooth profiles are destroyed, increasing sensitivity to pitting and tooth breakage.

Scoring Characteristics: As considered here, scoring is a welding and tearing action resulting from metal to metal contact, which removes material rapidly and con-

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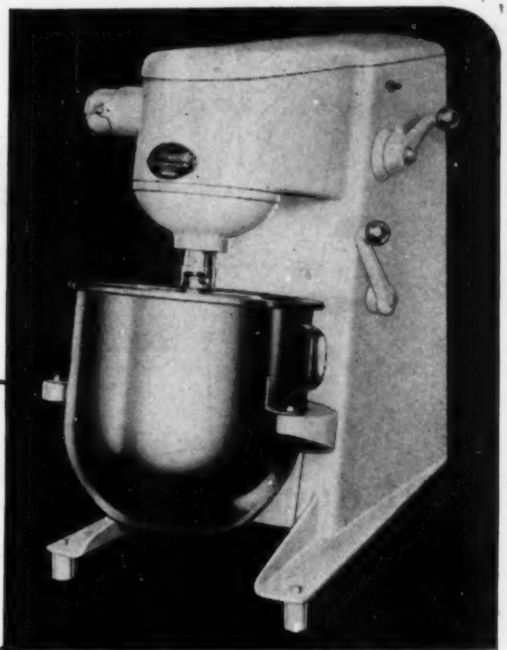

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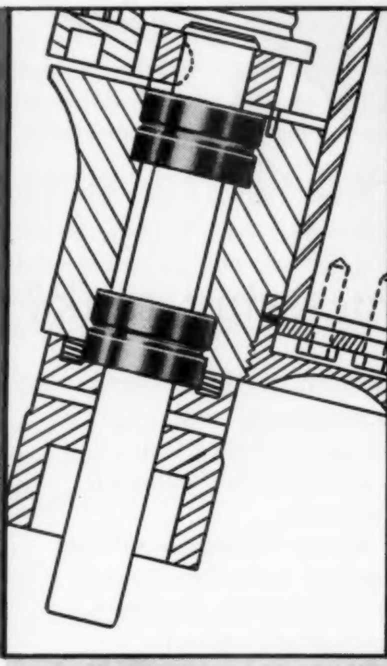
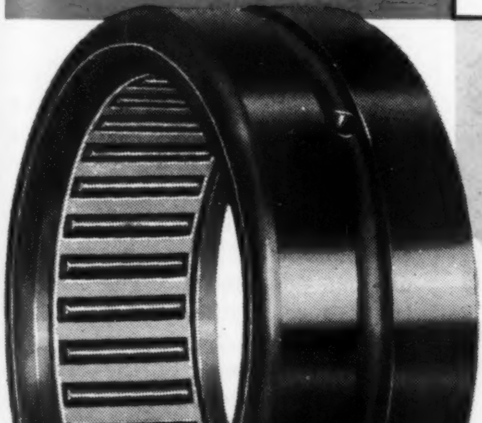


By tilting the beating mechanism in this 20-quart bench mixer, instead of the usual vertical position, engineers at Triumph Manufacturing Co., Cincinnati, O., have duplicated the same effect as whipping done by hand. Ordinarily, this would pose a real bearing problem—extreme compactness, high capacity, overhung loading, tilted vertical operation, variable speeds up to 475 r.p.m. Fortunately, Triumph engineers had been using Orange Cage Type Needle Bearings with great success on other vertical shafts in its line of beaters. Would they work on the tilted shaft? Yes! After long testing and field experience, equally fine performance is being obtained in the new tilted beater mixers.



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Design Abstracts

tinuously as long as the loads, speeds and other operating conditions remain the same. This type of scoring causes noise, aggravates pitting and eventually leads to complete destruction of the gear.

The most important reason why scoring phenomena have not been adequately expressed in an equation is our lack of specific knowledge of lubrication. Supporting data for previous equations have been meager, or have not covered a wide enough set of variables. This makes purely empirical criteria acceptable only within narrow limits of gear design.

Most of these equations include some relation between pressure and sliding velocity. Use of two variables simplifies calculation but causes inaccuracy. The following seven factors are considered as having significant effect on scoring resistance:

1. Pressure
2. Absolute surface velocities and resultant relative sliding velocity
3. Viscosity and composition of oil
4. Temperature of oil bath
5. Properties of material
6. Surface finish
7. Surface treatment

Since these factors are all common variables of modern gearing practice, an equation must take proper cognizance of them to be an effective design tool.

Scoring may be looked upon as a combination of two separate and distinct phenomena. The first is the failure of the oil film in the contact area. The second, which is visible evidence of the first, is the metallic welding and tearing that occurs. There is no contention here that full, thick film lubrication exists between gear teeth; in most cases it does not. Nevertheless, a supporting layer of oil in some form such as an absorbed film must exist and its destruction in the area of contact is almost instantaneous when final scoring occurs.

In an analysis of scoring, a pair of gear teeth may be likened to two rollers pressing against each other at some area of contact on their periphery, rotating in such a manner as to simulate the combined rolling and sliding of gear

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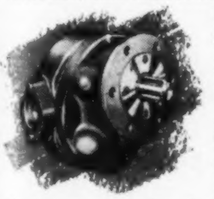

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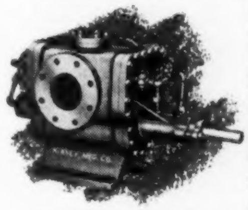

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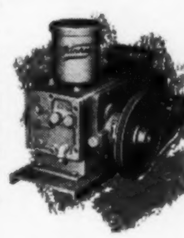

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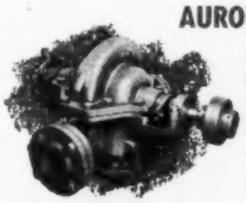

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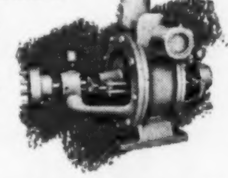

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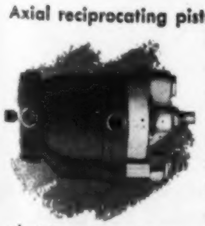

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teeth. Assuming the rollers are of a known material, having a known surface condition and finish, and the oil which is lubricating them is of known viscosity and quality, three variables remain which will score the rollers; namely, contact pressure, surface velocity and oil supply temperature.

Although surface temperature as a result of sliding and pressure has been implied in most formulas it is not adequately treated and the effect of bulk oil temperature has largely been ignored. Temperature alone may be the ultimate basis for the failure of straight mineral oils.

Surface Temperature: In 1937, Professor H. Blok developed a theoretical formula for instantaneous surface temperature that he calls a "temperature flash," which is based upon the conversion of friction energy to heat. Blok postulated that straight mineral oils have a critical temperature at which they would fail dependent only upon their viscosity. His postulate included the premise that such critical temperatures would limit the use of mineral oils on gears.

In Blok's derivation, band shaped contact pattern having a parabolic load distribution is assumed. The parabolic distribution is chosen in place of the actual elliptic one because of ease of calculation. The error is not significant for our purpose. A surface temperature pattern is formed in the contact area due to sliding as in *Fig.* 1. The instantaneous surface temperature change, T_2, is added to the bulk temperature of the material T_1, to form a final surface temperature, T_f. The peak of the temperature pattern lags the center line of the

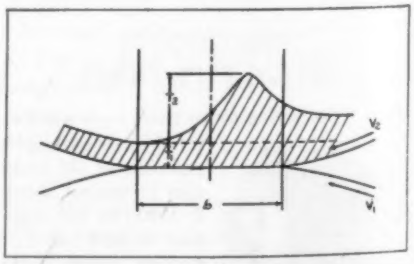


Fig. 1—Surface temperature pattern in the band of contact for gears in mesh


THE GEAR GRINDING MACHINE CO.
DETROIT, MICH.

contact band width, b, an amount dependent upon the surface velocities. This heat penetrates only a small distance into the material and the temperature quickly returns to the bulk temperature as contact passes. Blok's original equation, using American terminology, is

$$T_2 = \frac{K f W_n (V_1 - V_2)}{\sqrt{\frac{b}{2}}(C_1\sqrt{V_1} + C_2\sqrt{V_2})} \quad (1)$$

where K is a constant; f is the coefficient of friction; W_n is normal load per unit length; V_1 and V_2 are surface velocities; C_1 and C_2 are constants of the materials which include their thermal conductivities, specific heats, and densities; and b is the width of the band of contact as calculated by the Hertzian elasticity formula.

This equation includes three of the seven variables mentioned as having an effect on scoring: load, velocity and characteristics of the material. If the bulk temperature T_1 is included then

$$T_t = T_1 + T_2 \ldots\ldots \quad (2)$$

where T_t is the final temperature at the surface.

Characteristics of the material included in the formula are not meant to give a complete picture of the load carrying ability of mechanisms such as worm gears. The success of such gears is more dependent upon physical properties of the material, such as the ability to conform to a mating surface and possibly some factor such as compatability of materials.

By thermoelectric temperature measurements on spur gears Blok was able to prove the proportionality of the instantaneous temperature formula. In 1939, Blok, on an analysis of the Boerlage Four-Ball Extreme-Pressure Lubricant Tester, justified the application of the critical temperature hypothesis to that machine. In 1950, T. B. Lane showed that the critical temperature postulate worked on a simple two-ball scoring test machine. Attempted use of the hypothesis on gears, however, was not very successful.

Fundamental Principles: A cer-

Design Abstracts

tain basic error has been evident in previous investigations of scoring on gears. Scoring occurs in either or both the addendum and dedendum of the gear. Investigation, therefore, is somewhat confined to a period in the mesh of a gear tooth when two pairs of teeth are sharing the load, or the range of double tooth contact at the beginning and end of mesh. When considering the application of any formula to gears there is little justification in assuming that the pressure applied at the tips of the teeth depends only upon the applied tangential load or torque on the gears. Empirical formulas will almost always ignore load sharing in this range of double tooth contact. Thus such formulas cannot be expected to work satisfactorily for sets of gears which do not have exactly the same errors and geometrical characteristics.

A pair of gears having perfect involute profiles, perfect spacing, and perfectly uniform load distribution across the face of the teeth may be analyzed as follows. Considering the flexibility of the teeth, a load curve is obtained on a single pair of teeth as they go through mesh similar to the top diagram on *Fig.* 2. The smallest load exists at the start and end of contact since the combined deflection is greatest at the tips of the teeth.

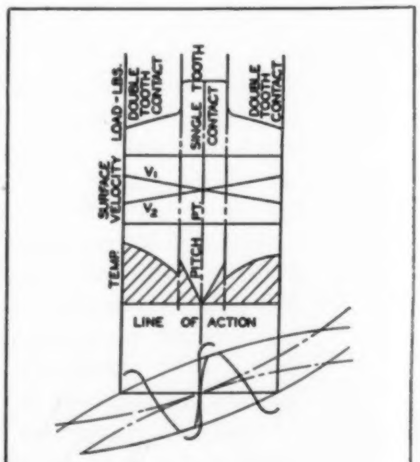


Fig. 2—Load, surface velocities and surface temperatures of mating gear teeth with perfect involute profiles and perfect spacing

The curves just below this on the same figure show the surface velocities of the two teeth as they pass through mesh. These velocities are instantaneous velocities of points tangent to the profiles at the points of contact. The difference of the two velocities is, of course, the sliding velocity.

With this information, assuming a constant coefficient of friction, the instantaneous surface temperature can be calculated for all points on the profiles of the mating teeth. The resultant temperature pattern is shown in the lower curve of *Fig.* 2. On each side of the pitch point the temperature increases rapidly to the ends of single tooth contact. A sudden drop in temperature occurs when two pairs of teeth share the load, then a much more gradual temperature rise takes place to the end of mesh. This latter portion of the instantaneous temperature curve almost flattens out near the tips of the teeth.

Sources of Error

There are several values in this analysis which are in need of refinement. The first is an assumed constant coefficient of friction. In Equation 1 the surface temperature is directly proportional to this factor. Many charts have been published showing the effect of sliding velocity on the coefficient of friction. Practically all of them are different and apparently dependent upon the test apparatus on which the data were obtained. Data taken on the coefficient of friction on actual gear teeth and on rollers show the value to be approximately constant. This work is certainly not conclusive and, because of dynamic problems of the test equipment above 360-fpm pitch line velocity, friction at speeds that normally occur in gearing could not be investigated.

The second possible source of error is the lack of dependable values on actual stiffness of the teeth. These values are very important when calculating the load distribution in the range of double tooth contact. Several investigators have published work on the subject. Each gives somewhat differ-

ent results and an analysis based upon them is questionable.

A third possible source of error exists in calculations of the width of the contact band, particularly near the tip of the tooth. Equation 1 shows that the temperature is inversely proportional to the square root of this band width. A wider band means a lower temperature flash. The calculations are based on the well known Hertz formula for this value, which was developed on rollers. Gear teeth are not rollers, and considerable error may exist near the tip of the tooth due to the lack of supporting material beyond the tip.

Test Results: These possible discrepancies have been discussed for the very pertinent reason that mathematically analysis does not agree exactly with test results. According to the lower curve of *Fig.* 2 the temperature appears to be higher at the tip of the tooth than at the end of single tooth contact. The calculated difference between these peaks is rarely more than 30 F. A very important observation from tests on 20-degree pressure angle gears is that the scoring failure is sensitive to the temperature at the highest point of single tooth contact if no involute modification such as tip relief is used.

Scoring tests were run normally with the gears in half mesh. A fairly definite line limits the bottom of the scored area. In more than 90 per cent of the gear scoring tests this line was found to correspond to the highest point of single tooth contact, or just short of the point where two pairs of teeth commence to share the load. It was found that removal of the portion of the tip of this tooth had no effect on the scoring resistance. Scoring started at the highest point of single tooth contact and was apparently dependent on conditions in that area.

On the assumption that this highest point of single tooth contact was normally the critical point of scoring for our 20-degree pressure angle test gears, values for total surface temperature were calculated in accordance with Blok's formula for this point on all

tests. Good results were found when considering only one design and batch of test gears, but changes in design and small differences in the machining of a new batch of gears—even with the same cutting tools—seemed to produce a different level of critical temperatures for the same oil.

Wide variations of surface finish are normally obtained in the shaving operation, unless this quality is carefully controlled. The surface finish values of the test gears ranged from approximately 17 to about 27 rms microinches. These values of surface finish were obtained after the gears were tested, in the area immediately below the start of scoring. The measurements therefore included the improvement of the surface due to running-in. Measurements of surface finish before the gears were run were found to be as high as 50 rms microinches. This surface finish reduced to the 27 rms microinch value within several hours of running. The important criterion is the surface finish of the gear after it has run-in rather than its original finish. No doubt, different machining practices and materials, producing or having the same initial surface, will reduce to different levels of roughness after running for only a short time.

The following formula was found to agree very well with the data:

$$T_t = \frac{T_1 + \dfrac{0.00317\,(W_n)\,(V_1 - V_2)}{\sqrt{\dfrac{b}{2}}\,(\sqrt{V_1} + \sqrt{V_2})}}{1 - \dfrac{S}{55}} \qquad \ldots\ldots(3)$$

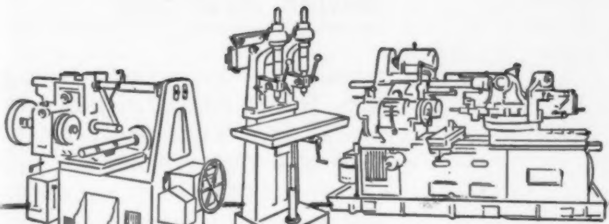
where S is the surface finish, rms microinches. The constant 0.00317 includes thermal properties of hard steel mating with hard steel and a constant coefficient of friction of 0.06. [More detailed information on the construction of this formula may be found in the original paper.]

As is shown in Equation 3, surface finish plays a very important part in the scoring resistance of gears. This formula has proved satisfactory for surface finishes between 7 and 27 rms microinches. Naturally, the formula becomes

questionable as the value of S approaches 55. However, as mentioned before, surface finish readings that are taken before the gears are run are worthless. On some tests where surfaces were purposely roughened by abrasive blasting, a few gears were scored during the run-in. Others which had undergone the same treatment, if able to get by the light run-in load, would withstand loads up to the estimated scoring load without failure. A new gear would either score within a few cycles or smooth over sufficiently in the same length of time to support the load. The first hour of operation is a critical period for scoring resistance.

Surface Treatment: The last of the seven factors affecting scoring is the influence of surface treatment. Some such treatments have shown the ability to permanently increase the scoring resistance of gears. Most surface treatments form a type of surface that assists break-in, such as a manganese-phosphate treatment with an etch on the basic material and a granular coating over this. Measurements of the surfaces after they have been run have indicated a decided improvement in the roughness, much more than is generally gained through the normal running-in procedure used on scoring tests. Soft metallic coatings such as those produced by copper or tin plating treatments no doubt tend to flow under operating temperatures and pressures, probably tending to fill the lower surfaces of the roughened area and making them support a larger than normal portion of the load, thus obtaining a better distribution of pressure over the contact pattern.

A note of caution should be mentioned about temperatures. The gear blank temperature is not always the same as the bulk oil temperature. Differences approaching 30 F are not uncommon, particularly where an externally cooled jet of oil is the only source of lubrication. In such cases the temperature of the gear blank may run significantly higher than the temperature of the oil jet. Meas-



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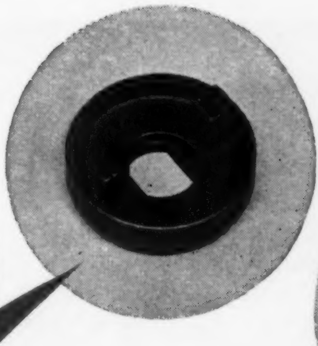

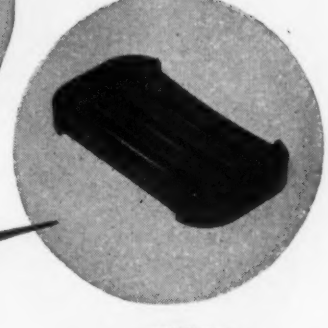

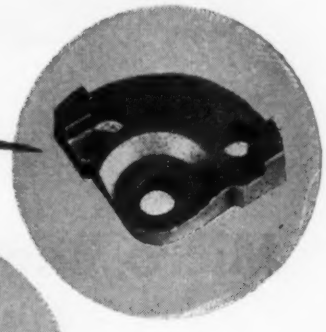

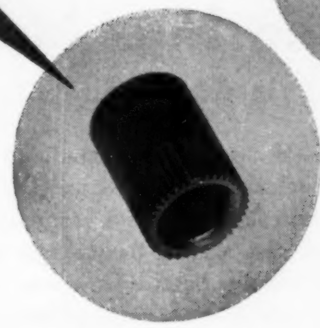

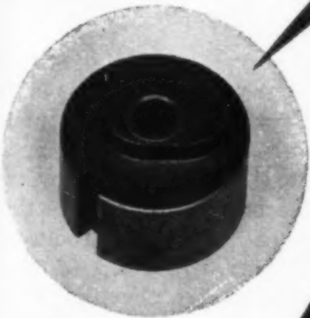

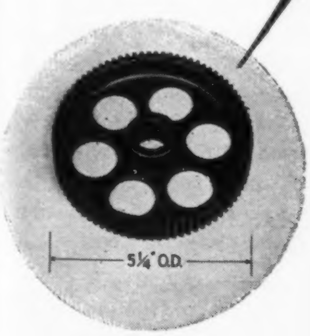

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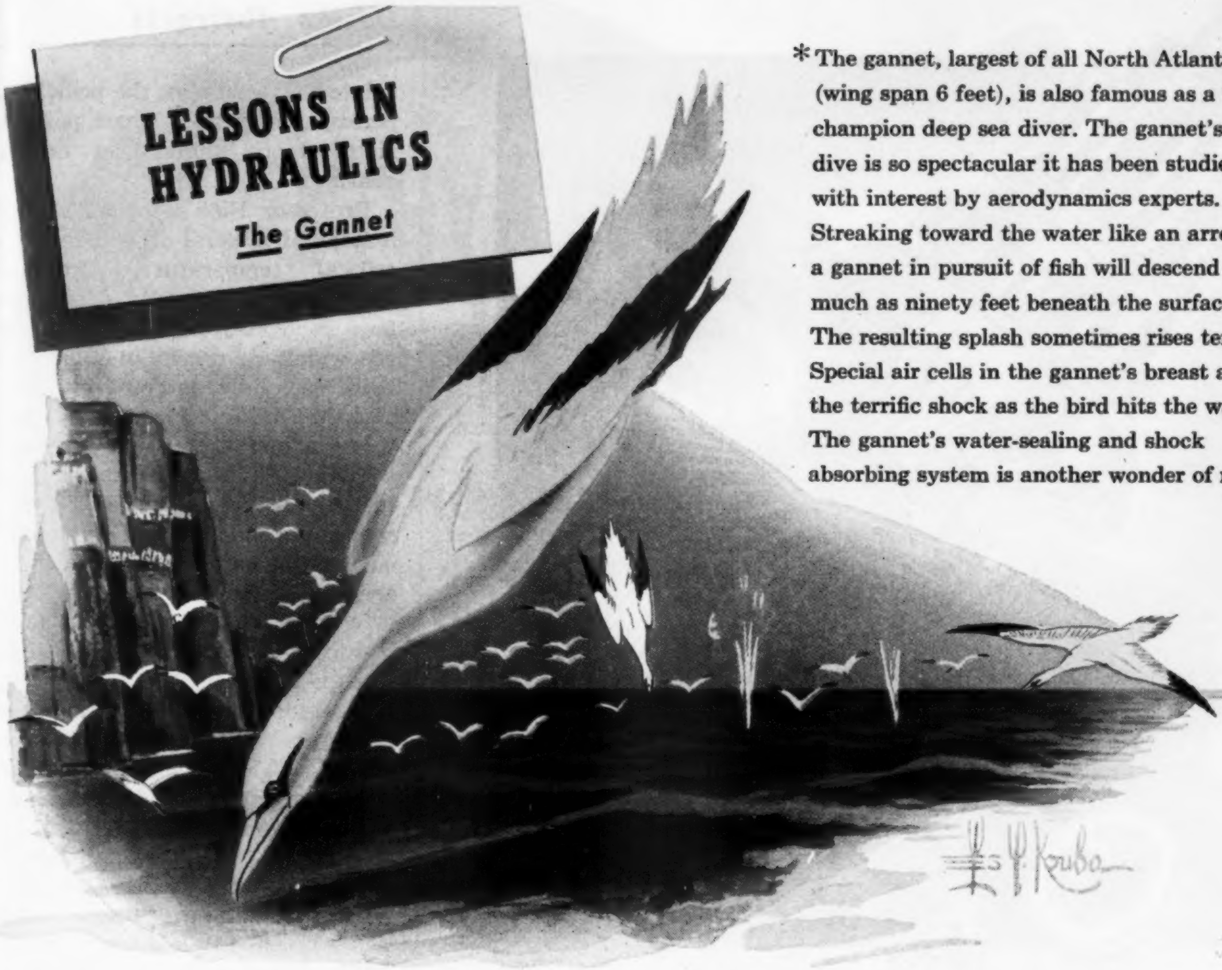
urements of gear blank temperatures of jet lubricated gears were taken from a thermocouple suspended in the oil stream being thrown off the gear. It was felt that this value was much closer to the true gear blank temperature. When splash lubrication is being used with no external oil cooler, the blank temperature is undoubtedly very close to bulk oil temperature.

Double Tooth Contact: Many scoring problems have been brought to the writer's attention that indicate the critical scoring point is not always the highest point of single tooth contact. The effect of profile modification, pressure angle and other factors of gear geometry can readily change the scoring picture. It is necessary that more than a superficial attempt be made to understand these variations.

A good example of scoring starting higher than the highest point of single tooth contact is found with high pressure angle gears. The higher the pressure angle the greater the stiffness of the teeth. This increase in stiffness produces a temperature at the tip of the tooth that is much higher than the temperature at the highest point of single tooth contact. When this difference becomes sufficiently great, scoring will start at a point during double tooth contact. If profiles are unmodified and scoring does not extend down to the highest point of single tooth contact, the probability of running into serious removal of material is small.

Summary: Modification in the form of tip relief is rather common practice today. Sometimes scoring troubles are a result of this modification. Suppose that the tip of the mating gear is relieved in the case of a long addendum pinion. This throws more load on the addendum of the pinion thus raising the surface temperature in that region, possibly above the critical temperature of the oil. Therefore, tip relief can either help or aggravate the scoring resistance of gears. Improvement caused by tip relief can only be to a certain extent and the amount may be cal-



* The gannet, largest of all North Atlantic birds (wing span 6 feet), is also famous as a world champion deep sea diver. The gannet's dive is so spectacular it has been studied with interest by aerodynamics experts. Streaking toward the water like an arrow, a gannet in pursuit of fish will descend as much as ninety feet beneath the surface. The resulting splash sometimes rises ten feet. Special air cells in the gannet's breast absorb the terrific shock as the bird hits the water. The gannet's water-sealing and shock absorbing system is another wonder of nature.

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Design Abstracts

culated by assuming the peak temperature is at the highest point of single tooth contact in normal gearing.

Professor Blok's postulate, that a straight mineral oil will fail at a critical temperature dependent only upon its viscosity grade, has been confirmed on machines which are widely divergent in design, on balls, on rollers, and on gears. Considering this correlation, the postulate shows promise of becoming a reliable lubrication design tool. However, the proper use of such a tool is dependent upon our knowledge of the many factors which enter into it and our ability to properly apply them to our problems. Purely empirical formulas, although necessary as a temporary design tool, are rarely expandable and cannot be trusted for advanced designs. The approach which attempts to deal with more fundamental concepts is more desirable because of its probable expandability. Even with only fair initial applicability it will form a base which is generally added to instead of discarded with the advent of more stringent design requirements.

From a paper entitled "A New Look at the Scoring Phenomena of Gears" presented at the AGMA Semiannual Meeting in Chicago, Ill., October, 1952.

Controlling Wound-Rotor Motors

By W. J. Heacock
Electrical Engineer
Link-Belt Co.
Chicago, Ill.

CONTACTOR control of wound-rotor motors has been utilized in machinery applications for a number of years. It provides an average torque during acceleration by progressively cutting out varying degrees of resistance in the secondary circuit of the motor. The acceleration from this type of control is by no means uniform, and the peak torques contribute shock to the machinery. If an upper limit of torque is established for protection of the drives, then the time

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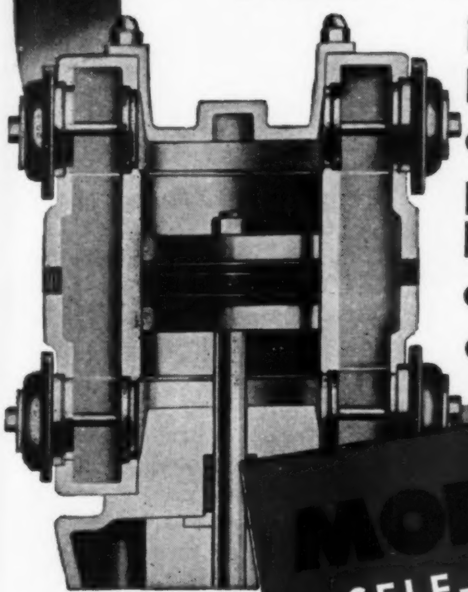
in compressors handling foods, chemicals, plastics, beverages, oxygen, etc.



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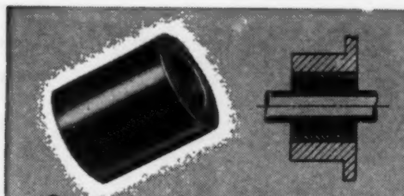

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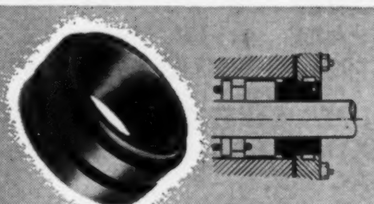

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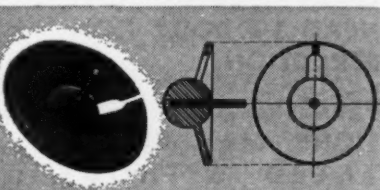

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required for acceleration is unnecessarily long because the motor torque falls off as the motor increases in speed on each point, thus leaving a lower torque margin for acceleration.

How time is lost during acceleration due to this type of control is shown in *Fig.* 1. The wavy line *OB* indicates the time required for acceleration under a nonuniform torque application as compared to the shorter time *OA* under a uniform torque equal to the maximum allowable.

Although this ideal curve cannot be exactly duplicated, it is closely approached by a system which Link-Belt has been using for the past few years with success. The principles involved, taken separately, are not new, but the combination employed is believed to be a new concept.

First step in the design of this control is to add sufficient resistance to the secondary circuit to cause the motor to develop maximum torque at standstill. Usually the right amount of resistance is from 17 to 20 per cent ohms, one hundred per cent ohms being the value required to give rated full-load torque at standstill.

The maximum torque of most wound-rotor motors is generally high, usually too high for the drive. In the conventional method of control, the torque is reduced by adding more resistance. It takes about 67 per cent ohms to develop 150 per cent torque at standstill. In the improved system, the torque is reduced to the desired value, not by increasing the secondary resistance,

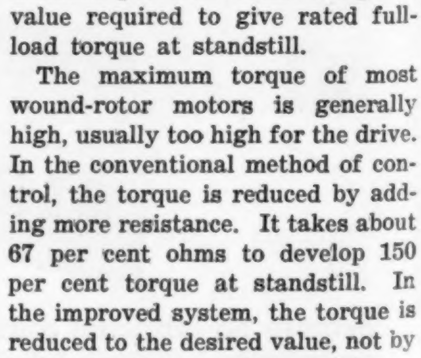

Fig. 1—Acceleration time for conventional contactor control compared to ideal. Sawtooth curve at far right is acceleration curve for conventional eight-point control


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but by introducing some resistance in the primary circuit. Since the motor torque is proportional to the square of the impressed voltage, it does not require a great amount of resistance in the primary circuit to reduce the torque to the desired value.

The voltage at the motor terminals has been reduced by virtue of the voltage drop in the primary resistance due to the flow of current in the primary circuit. As the motor increases in speed the primary current decreases, which results in less voltage drop across the primary resistance which, in turn, increases the voltage at the motor stator terminals. The torque of the motor would also drop if the terminal voltage remained constant but since the voltage is increasing the expected torque loss is compensated for and the torque remains at a constant or slightly increasing value. This, then, gives essentially constant torque up to something over seventy per cent of the motor speed. The remaining points are about the same as with the conventional control.

The complete power circuit and resulting speed-torque characteristics are shown in *Fig.* 2. Only three contactors are required but four points are obtained. This is accomplished by using number two contactor twice. A conventional control with seven or eight contactors would not give a comparable speed-torque curve. It would give an average torque available for acceleration somewhat less, therefore requiring more time for acceleration.

In an existing application, where a 150 hp motor drives a heavy iner-

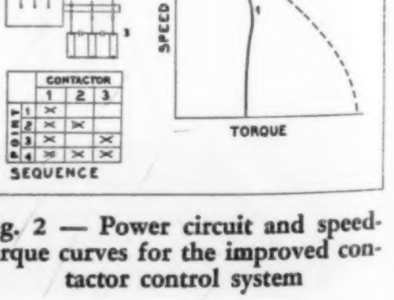


Fig. 2 — Power circuit and speed-torque curves for the improved contactor control system

tia load by means of V-belts, good comparison of this type of control with the conventional contactor control was obtained. Even though seven accelerating contactors plus the line contactor were figured to minimize the spread between maximum and minimum torques, the time for acceleration was three minutes. With the new system time of acceleration was cut to two minutes and only three contactors were employed.

No attempt has been made here to show the control methods utilized for switching the resistors in and out. These vary with the application but are relatively simple. Enough of these control systems are now in successful operation to demonstrate it is not just a novelty, but rather that it gives a decided improvement in performance with smoother acceleration and maximum utilization of the drive which is employed.

From a paper entitled "An Improved Control for Wound-Rotor Motors" presented at the Fourth Biennial Materials Handling Conference sponsored by Westinghouse Electric Corp. in Buffalo, N. Y., October 1952.

Applying Power Hydraulics

By R. H. Fritzges
Mack Mfg. Corp.
Allentown, Pa.

HYDRAULIC actuation of the steering mechanisms of passenger cars has in recent months captured the interest of automotive designers as well as the motoring conscious American public. Equally interesting, and perhaps more important, has been the utilization of hydraulic power on large buses not only for steering, but for all the other auxiliaries requiring a power source. A partial list of these power-operated control devices would include brakes, throttle, doors, windshield wipers, and the units that control shifting of the automatic transmission from torque converter to direct drive.

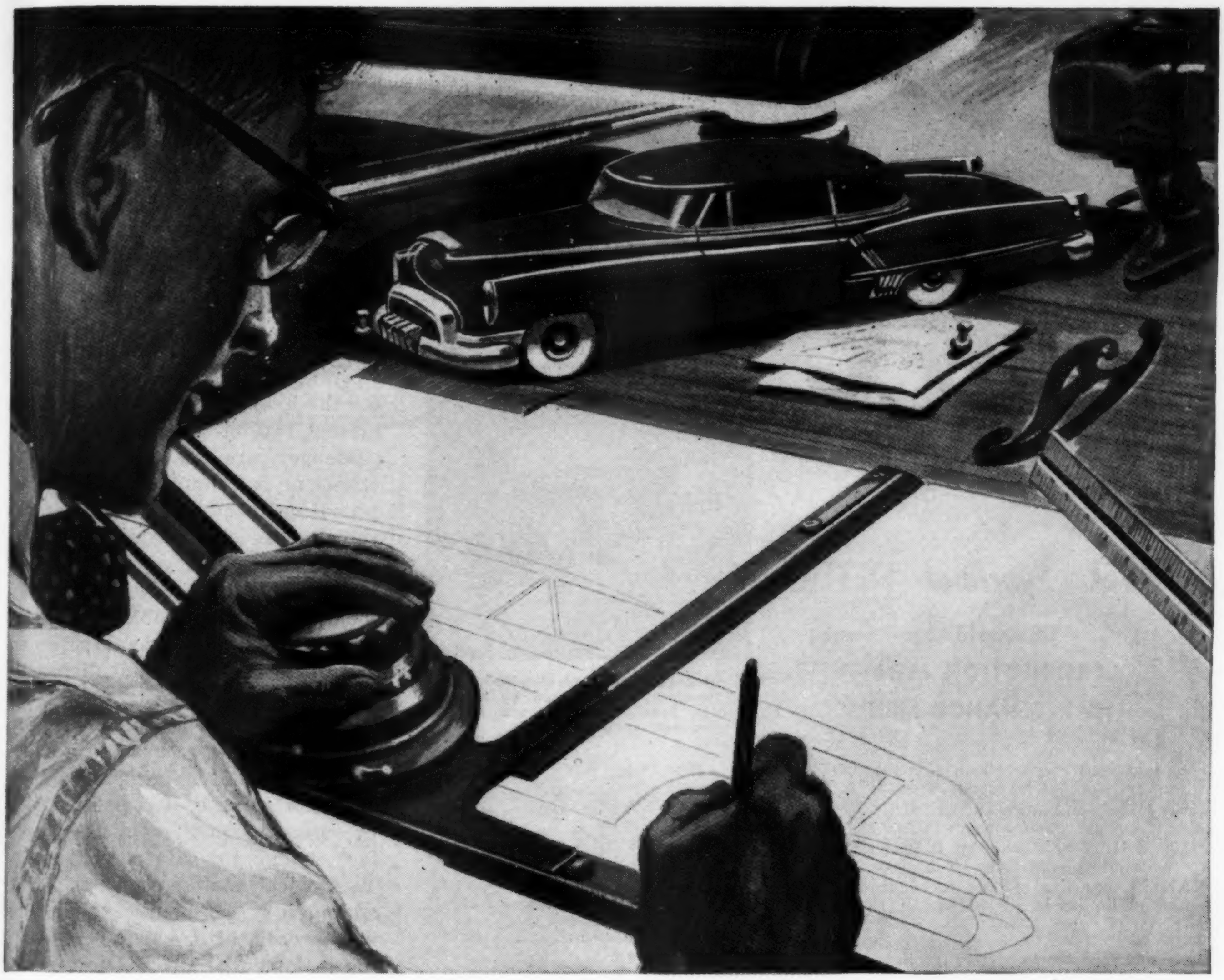
There are many reasons for selecting a hydraulic fluid as the working medium on automotive

equipment, and a partial list of such reasons would include first of all, weight and size reduction. In addition, the hydraulic medium is virtually incompressible and, therefore positive and swift in action; the equipment used in hydraulic systems is self-lubricated and possesses an inherent protection from overloading, and can be applied to a competitive or comparable cost to that of present pneumatic equipment.

Power hydraulics received much attention and was used extensively in the last World War on tanks, large trucks, and especially in aircraft where it efficiently performed many important and varied functions. Also, the machine tool industry has found hydraulic power a willing and able servant in the huge presses and similar equipment so necessary to our mass-production methods. The benefits listed earlier were instrumental in the selection of hydraulics as the working medium in these fields and a bonus has been realized in most instances due to reduced maintenance costs.

A bus can be equipped with the hydraulic system without any great changes in manufacturing methods or techniques as used on air systems. A weight saving of 150 pounds has been recorded with the hydraulic system and this does not include the bonus of power steering, which counts for about 90 pounds more.

Future Possibilities: As was indicated previously, a full pressure hydraulic system provides many interesting possibilities for all kinds of automotive applications. A relatively new field in this broad development is that of the hydraulic starter which is being used experimentally on large diesel truck engines. This starting system which uses a pump, accumulator, and reservoir in addition to a hydraulic starting motor, could possibly be included on a truck or bus already equipped with a full pressure system and thus utilize a common pump, reservoir, and accumulator. Of course, once a power source is established on any vehicle, all of the controlled devices could be pow-



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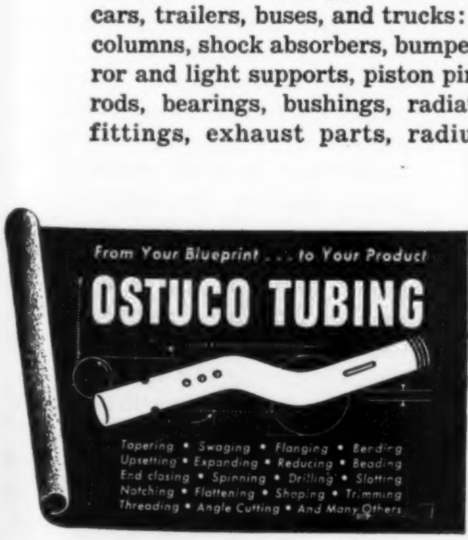

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ered by the common working medium. The mechanism that works dump bodies, power operated tail gates, winches, and similar functions could all be operated by a full pressure hydraulic system.

Safety Is Increased

Naturally, the hydraulic power steering is the major justification for the hydraulic system, and if its present rate of increased usage on passenger cars can be considered a criterion, it cannot be long postponed on heavy-duty commercial vehicles. A lot in physical exertion is required to steer a vehicle which may have a front axle load of $3\frac{1}{2}$ tons. This latter problem has been recognized by bus manufacturers and up to a certain limit, help can be offered by increasing the steering ratio. But the limit is rather quickly reached because to get effortless manual steering on a bus, the ratio would be at a point where the driver couldn't spin the wheel fast enough for safe maneuvering. The logical answer seems to be in adding a power source to do the heavy work, after which the steering ratio can be lowered to a point where fast, positive steering control is assured. If it is then conceded that this reduction in steering effort will result in a similar reduction in driver fatigue, it would appear that a very favorable step has been taken towards increased efficiency and greater safety on streets and highways.

The steering system, of course, is thought of here as a part of a full hydraulic system and present indications are favorable to its use on all types of automotive vehicles. It is generally agreed that vehicle designers are continually striving to make their products easier to operate, cheaper to maintain and safer to use. It is well considered that the effective use of hydraulic power may result in worthwhile gains in all three functions—operational ease, less maintenance, and above all, increased safety.

From a paper entitled "Total Control with Power Hydraulics" presented at the SAE National Transportation Meeting in Pittsburgh, Pa., October 1952.

New Machines

Communication

Television Camera: Portable; completely self contained. Useful for remote control, hazardous inspection and checking operations. Picture can be picked up by any standard television receiver. Daylight or normal room light are usually sufficient for clear transmission and reception. Electronic film finder and all camera controls are readily accessible. Installation can be made with a cable connection between camera and receiver, each connected to a standard power source. Mounts on any standard camera tripod or can be permanently mounted to fit conditions of installation. Size of unit, 14 in. long, 9⅜ in. high, 4⅝ in. wide. *Dage Electronics Corp., Beech Grove, Ind.*

Transcription Unit: Model 2195 includes microphone and microphone-or-instrument inputs in addition to built-in phonograph equipment. Microphone, instrument and phonograph inputs have separate volume controls. Tone controls affect only phonograph and micro-instrument inputs; musical background can be controlled without affecting microphone announcements. Seven-tube amplifier with push-pull beam power output tubes and a 16-in. tone arm plays records from 6 to 16 in. at all speeds from approximately 30 to 80 rpm. A 12-in. heavy-duty speaker mounted in the removable lid is equipped with a 25-ft SV cable and plug. Power output of unit is 10 w. Size, 8½ in. deep, 14 in. wide, 15 in. high. *Bell Sound Systems Inc., Columbus, O.*

Domestic

Refrigerators: Two 2-ft wide models designed for apartment and limited space installations. Each has aluminum ice trays, rust-resistant anodized aluminum shelves, three door shelves, full-width vegetable drawer, automatic interior light, adjustable temperature control and full-width freezer com-

New Machines

partment which holds up to 22 lb of frozen food. Model LB-76 has 7.6 cu ft capacity; defrosts manually. Model LC-70, with 7 cu ft capacity, features cold air circulation and automatic defrosting action. This model has a sloping aluminum baffle below the full-width evaporator which causes cold air to circulate down the back and around to the top of the cabinet. For defrosting, a sealed heating unit built into the evaporator is controlled automatically by a frost-limiting device. Water from defrosting is channeled into a glass receptacle on the top shelf. *Major Appliance Div., General Electric Co., Louisville, Ky.*

Food Waste Disposer: Fastens to sink bowl with three bolts. Neoprene gaskets at drain opening and tailpiece absorb vibration. Tailpiece, made of tubing rather than cast metal, can be swiveled or turned to most convenient position for connection to waste line. Self-reversing motor action; a toggle switch for activating the disposer may be installed on the wall. Drain opening is 6½ in. below the bottom of the sink bowl, permitting installation to existing drain lines in the wall. Overall length of unit is 14 in. *Mullins Mfg. Corp., Warren, O.*

Heating and Ventilating

Room Air Conditioners: Thirteen models in sizes ranging from ⅓ to 2 hp. All have hermetically-sealed refrigeration circuits and are designed for quiet operation. Portable console style conditioner can be installed in any window which opens. Reverse-cycle units, which heat as well as cool, are available in window and console types in ¾ and 1-hp sizes. Modulation control, which prevents overcooling, is featured with the reverse-cycle units and is also included in a number of other models. *York Corp., York, Pa.*

Air Circulator: Portable 18-in. model designed for offices, stores and homes. Adjustable in height from 28 to 48 in., floor to center of blades. Vertical tilt adjustment permits directing air flow within an arc of 180 degrees. Has two-

New Machines

speed capacitor motor; delivers 4350 cfm of air at high speed. *Emerson Electric Mfg. Co., St. Louis, Mo.*

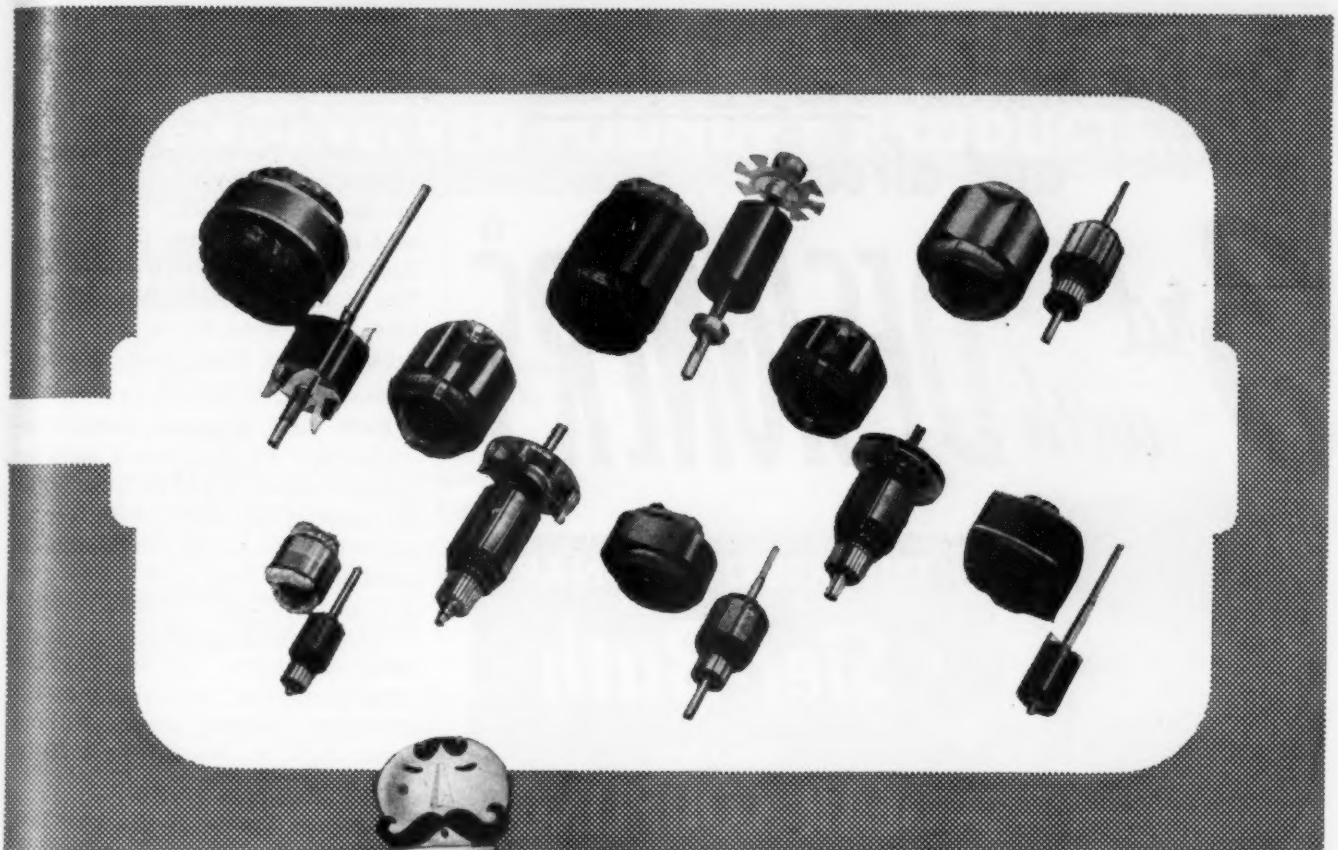
Water Chiller: Designed to provide a single-unit source of large quantities of cold water for air conditioning and refrigeration systems requiring between 10 and 50 tons of cooling. Recommended for use in offices, stores, hotels, shopping centers, hospitals and apartment buildings. Self contained; combines compressor, condenser, liquid cooler, controls, motor and accessories in a single factory-assembled and tested unit which requires only plumbing and electrical connections. Heart of the unit is a multistep reciprocating compressor. Power and water consumption is automatically adjusted to the demand for cooling. Compressor's plastic-coated valves, lubrication system, shaft seal and other features are designed for minimum maintenance and servicing. *The Trane Co., La Crosse, Wis.*

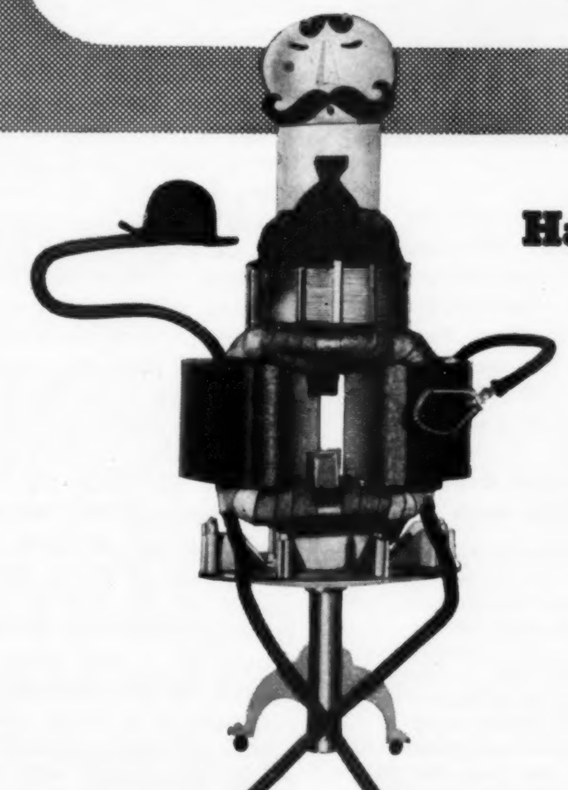
Reversible Window Fans: Switch controls inward or outward movement of air. Can be installed in either the lower or upper portion of double-hung windows; are available in 24 or 30-in. sizes. Two-speed, split-phase motors are resiliently mounted for quiet operation. On high speed the 24-in. fan delivers 5000 cfm of air; the 30-in. model delivers 6700 cfm. *Emerson Electric Mfg. Co., St. Louis, Mo.*

Materials Handling

Fork Lift Straddle Truck: Electrically powered; available in 2000 and 3000-lb models. Designed for right-angle tiering in 5 to 6-ft wide aisles and for operating in elevators and on flooring that will not sustain heavier rider type electric lift trucks. Has three forward and three reverse speeds. Can lift loads as high as 133 in. *Yale & Towne Mfg. Co., Philadelphia, Pa.*

Hand Truck-Trailer: Model A-310-326M has capacity of 4000 lb, is designed to handle palletized loads. Equipped with simple brake mechanism to permit easy manual control of truck and load when operating on grades. Trailer brakes





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are of external contracting type and are applied by depressing a hand-operated lever located on the caster-end pipe rack. Couplers at each end of unit equip it for trailer duty in trackless train operations. Has molded-on solid rubber wheels. Size, 36 in. wide, 62 in. long; flush deck is 13 in. from floor. *Mercury Mfg. Co., Chicago, Ill.*

Fork Lift Trucks: Diesel-powered Model FTD50-24 and gasoline powered Model FT50-24 have 5000-lb capacity at 24-in. load center from heel to forks. Available in standard lift heights of 72, 84, 108, 114 and 120 in. Feature single-lever full-range gear shift, center-point automotive type steering, hydraulic brakes, self-aligning lift cylinder. Overall length without forks is 88 in.; wheel base is 50 in. *The Buda Co., Harvey, Ill.*

Towing Tractor: Clarktor-120, designed for aircraft manufacturers, commercial airlines and heavy industry. Has drawbar full rating of 12,000 lb; powered by 16A Chrysler engine through fluid coupling. Tractor has four speeds forward, with a maximum of 16.3 mph; one reverse speed of 2.17 mph. Rear axle embodies double-reduction gearing, with the final reduction of the planetary type mounted in the wheels. Drive shaft extends through the tube that supports wheel bearings and transmits torque to a set of three planetary gears, each of which transmits its portion of the load to the wheel. Reaction from these gears is transmitted to a ring gear which is splined to the housing. Machine can turn in a 127-in. radius. Low silhouette provides good visibility. *Clark Equipment Co., Industrial Truck Div., Battle Creek, Mich.*

Fork Lift Truck: 4000-lb capacity "Octopus" handles ordinary pallet loads; works as a side-shifting truck close to columns and walls; hauls and stacks drums and paper rolls and dumps, hauls and stacks tote-box loads. Can also pick up a variety of pallet sizes after automatically adjusting for proper fork spacing. Drumloads of bulk material are emptied by hydraulically tilting drums in a vertical plane. Bales of cotton and

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PROBLEM:

1. This part for an assembly line conveyor was not being produced economically as a weldment.
2. The part must withstand constant shock and rough handling of the upper hinge member.

SOLUTION:

The proven principles of SUPERIOR ENGINEERED FOUNDRY DESIGN were applied in redesigning the part instead of merely duplicating the original weldment design as a steel casting. Machining of the part was done by Superior's machine shop.

RESULT:

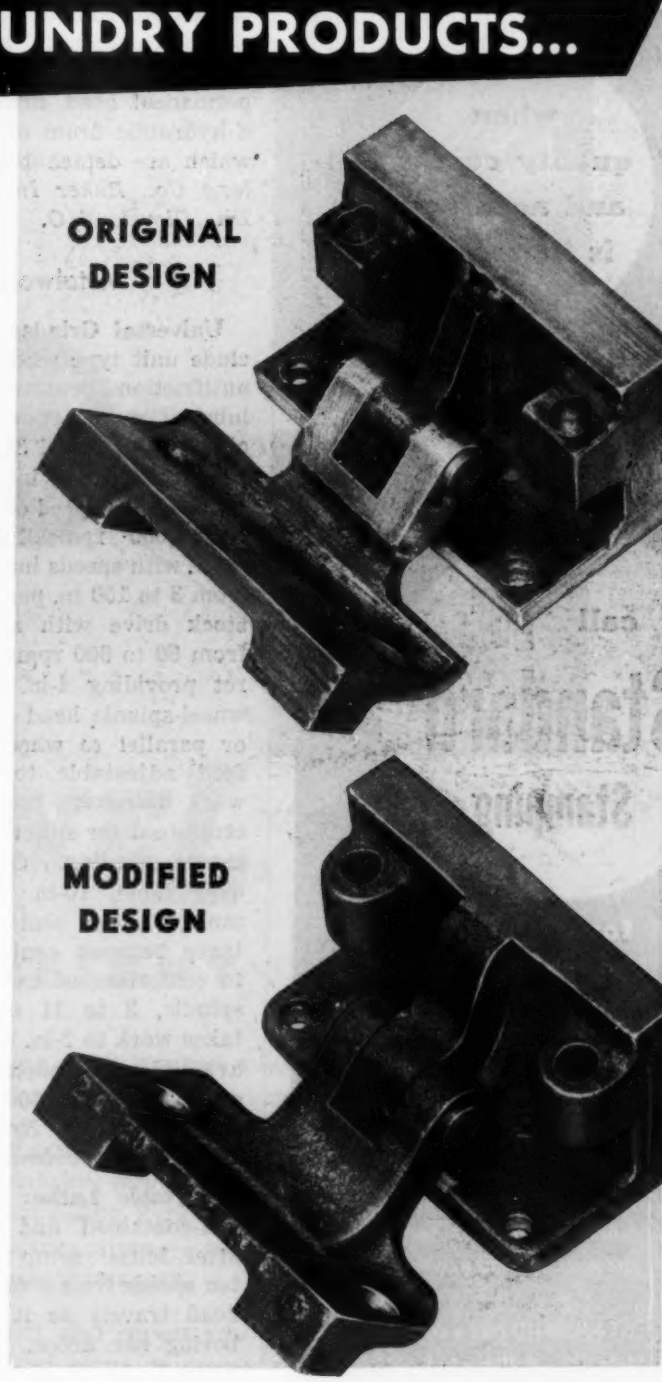
50.2% SAVINGS

1. The steel casting costs less to produce and less machining is required because the hinge pin hole is cored. Superior delivers complete, machined units, ready for installation.
2. The part has the steel casting's inherent strength for continuous hard use plus a weight reduction of 7.2%.

Total Cost of Part Reduced 50.2%.

YOU, TOO, CAN GET SAVINGS LIKE THESE BY CONSULTING SUPERIOR'S PRODUCT DEVELOPMENT SERVICE.

Remember...you benefit through correct design. If it can be cast, Superior's service develops the best design in which to cast it. If it shouldn't be cast, Superior's service develops the reasons why.



Courtesy of Fabricated Steel Products of Indiana, Inc.

Make your parts Superior Engineered Foundry Products . . . steel castings to 30,000 pounds . . . malleable iron castings to 300 pounds.

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ENGINEERED FOUNDRY PRODUCTS

Since 1916

SUPERIOR STEEL AND MALLEABLE CASTINGS CO.

BENTON HARBOR, MICHIGAN, U. S. A.

crude rubber are lifted with a spike-clamp attachment. Uses three other attachments — a revolving semi-permanent head, drum clamps and a hydraulic drum up-ender—all of which are detachable. *Baker-Raulang Co., Baker Industrial Truck Div., Cleveland, O.*

Metalworking

Universal Grinder: Features include unit type wheel spindle with antifriction bearings and sealed lubrication; practical wheel speeds of 2500, 3000 and 3600 rpm; independently driven internal grinding spindle with speed of 16,500, 24,000 or 35,000 rpm; hydraulic table drive with speeds infinitely variable from 3 to 150 in. per minute; headstock drive with speeds variable from 60 to 600 rpm; universal turret providing 4-in. adjustment of wheel-spindle head at right angles or parallel to wheel slide; cross-feed adjustable to 0.0001-in. on work diameter; positive stops on cross-feed for either external or internal grinding. Capacity: swing over table, 10-in. diameter; distance between centers, 20 in.; distance between centerline of work to centerline of external grinding spindle, 2 to 11 in.; center rest takes work to 2-in. diameter. Also available are Models No. 2, 3 and 4 with 30, 40 and 60-in. distance between centers. *Brown & Sharpe Mfg. Co., Providence, R. I.*

Portable Lathe: Model B-23 is self-contained and semiautomatic after initial setup is made. Has ten speeds from 0 to 45 rpm. Lathe head travels as it revolves on a boring bar arbor. Tool arm, adjustable for diameters from 7½ to 20 in., remains fixed in relation to head. Design permits either external or internal machining of circular workpieces up to 20-in. diameter. *Barrett Equipment Co., Industrial Div., St. Louis, Mo.*

Bonding Machine: Produces bond between lead and steel. Steel is first prepared by leadizing; then lead is applied. Will bond a strip of lead up to 12 in. wide by ¼-in. thick to prepared steel. Width and thickness of cladding is controlled by regulating machine's speed and by spreading or narrowing distance

New Machines

between its two cladding heads. Lead can be carried by machine in tubular feeds at the front, or a strip of lead can be laid down under the machine. Cladding heads melt this lead and bond it as the machine automatically moves across the plate. Unit can run free, directly on the steel, or operate on a track. Can also operate in a tank shell if bottom carriage is removed, reversed and mounted on the top of the machine so it runs on an aluminum or magnesium beam. *Knapp Mills Inc., New York, N. Y.*

Internal Gear Finisher: Model 870-C for internal spur and helical gears from 4-in. pitch diameter to 12-in. OD. Smaller sizes can be accommodated with special cutter spindle. Has heavy-duty dovetailed slide suspension for cutter spindle; cone-drive gearing in cutter spindle reciprocating drive mechanisms; vernier scales and dial indicator that reduce setup time; easily accessible change gears and controls; sealed electrical panels; adjustable power-driven knee; one-shot lubricating system for ways, knee and rotating parts; pressure lubricating fittings on cutter and work spindle. Uses cutters having pitch diameters from 3 to 6 in. Maximum stroke of head is 5 in. Maximum gear face width is 4¾ in.; coarsest recommended work pitch is 5 diametral pitch. Powered by fan-cooled 3-hp main drive motor, ½-hp coolant pump motor, ½-hp head movement motor and ⅓-hp infeed motor. Size: 45⅝ in. wide, 62¼ in. deep, 84⅛ in. high; weighs 11,000 lb. *Michigan Tool Co., Detroit, Mich.*

Lathe Attachment: Turnomat has capacity up to 1¾ in. bar stock, with four interchangeable heads which permit rapid change from 1/32 to 1¾-in. stock. Can turn to small diameters in one plunge cut. Also can taper, knurl, center, form, thread, drill and ream. *Turnomat Co. Inc., Brockport, N. Y.*

Multiple Drill: Special machine drills thirty-two 3/16-in. holes in a steel part at the rate of approximately 120 pieces per hour. Incorporates eight Model HH drilling units, electrically interlocked with automatic, hydraulically operated

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clamping and indexing. Operator places part on a fixture and presses switch. Part is automatically clamped and eight holes are drilled simultaneously at each of four indexes. Part unclamps automatically. *Govro-Nelson Co., Detroit, Mich.*

Die Tryout Press: Capacity, 100 tons. Has 6-in. stroke; operates at 50 strokes per minute. Head can be released and rotated to any point up to 240 degrees; operator can barber, spot, shear, fit and finish both punch and die without removing die from press. Drive mechanism is housed in base and pulls head down. Electrically controlled clutch permits inching, single stroke, continuous operation and forward or reverse operation. Floor space required, 75 by 84 in.; height, 82 in. *Alpha Tool Works, Detroit, Mich.*

Centerless Grinder: Model TG-123 handles 1/16 to 1-in. diameter straight or contour work. Tolerances to 0.0003-in. can be held. Infinitely variable speed drive provides regulating wheel spindle speed range of 30-480 rpm. Grinding wheel spindle is mounted in preloaded, permanently lubricated superprecision ball bearings. Coolant tank is a separate unit mounted on large casters. Floor space requirement of machine alone is 23 by 36 in. *Royal Master Metal Products Co., Riverdale, N. J.*

Hydraulic Tracing Attachment: For use on any Jones & Lamson ram or 7A saddle universal bar or chucking machine for general tracing, multistep shaft and contour work. Powered by differential cylinder. Tool-carrying slide is mounted on a 45-degree angle with the spindle axis. Cylinder stylus and cutting tool are mounted on the back of the cross slide on a bridge type carriage. Attachment can be used for turning tapers; also provides facilities of a turret lathe on a profiling lathe. *Jones & Lamson Machine Co., Springfield, Vt.*

Flame Cutter: Three-spindle machine for producing gears and sprockets. Jury spindle attachment makes possible production of sprockets or gears larger than the machine itself. Multiple-spindle design permits automatic production


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You can realize startling cost savings when you use Rockrite® cylinder finish steel tubing for hydraulic cylinders, shock absorbers and similar parts. ● The inside is smooth, scratch-free—a surface just right for a piston with soft packing. Smooth bore surface plus exceptionally close dimensional tolerances means that this tubing can be cut-to-length and used as is. It is easy to make standard parts that are interchangeable. Slash your costs by using Rockrite cylinder finish tubing. ● Ask for Bulletin R-7 giving more data.
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of three gears or sprockets at a time in diameters up to 15 in.; two at a time up to 25-in. diameter. Automatically produces over 300 shapes and sizes of teeth, 5 to 200 teeth per wheel, in 3 to 108-in. diameters. *Cogmatic Co., Milwaukee, Wis.*

Welding Unit: Consists of work-clamping jig, power-operated welding head carriage and Heliarc welding equipment. Carriage is adjustable for welding speeds of 4 to 60 rpm on its horizontal track. Jig's end opening or gap permits welding all longitudinal seams of any formed item. Accommodates stock ranging from 28-gage sheets to ½-in. plates and is adjustable in pressure from 5 to 60 lb, depending upon material. Can be converted to Heliarc spot welding, producing welds capable of withstanding 300 to 900 psi pressure. *McHale Mfg. Co., Los Angeles, Calif.*

Pedestal Tool Grinder: Available with either 8-in. grinding wheels and ½-hp motor or with 10-in. wheels and ¾-hp motor. Motor is in base instead of between grinding wheels, which provides large work space around each wheel. Quiet, vibration-free running is accomplished with sealed ball bearings and fully enclosed V-belt drive. Large shields provide optical protection at each wheel; operator need not raise shield to see work. Two concealed lamps in each shield provide light for freehand precision grinding. Size, 49½ in. high, 18½ in. wide, 20½ in. deep. *South Bend Lathe Works, South Bend, Ind.*

Packaging

Heat Sealer: Automatically forms two sheets of heat sealing material around products, seals the four sides, cuts off and delivers a completely sealed package. Makes packages from 1 in. square by ½-in. thick to 6 in. square from materials such as cellophane, foil, glassine, heat-sealing kraft, etc. Speed, up to 40 packages per minute when products are hand fed. Attachments, including belt, turret, hopper and other units available for automatic feeding. Powered by ¼-hp motor; has thermostatically controlled heat. Size, 2

ft wide, 3 ft long, 3 ft high; weight, 500 lb. *Pak-Rapid Inc., Philadelphia, Pa.*

Stapling Machine: "Staple King," designed for simultaneous closing of tops and bottoms of center slotted cartons and partial or full overlap cartons. Production rate is more than 350 per hour. Retractable anvil feature permits closure from the outside after cartons are filled. Machine is air operated and semiautomatic. Stapling heads are operated by a foot pedal. Can be set for concealed stapling—staples are clinched approximately two-thirds of the way through the carton wall. Adjustable roller table guides cartons toward stapling heads. Machine automatically accommodates cartons up to 5½ in. high; an adjustment sets machine when variance is greater than 5½ in. *International Staple & Machine Co., Herrin, Ill.*

Portable Tools

Band Saw Blade Welder: Butt welds all blades from 1/16 to ¾-in.; permits range of work from intricate internal tool and die to power cut-off saws. Has built-in grinder to remove flash from the weld, as well as a double gage for checking thickness of weld on flat saws. Portable, but has removable back plate to facilitate permanent installation. Size, 7¾ by 12 by 7 in.; weight, 31 lb. *Brennen Mfg. Co., Brooklyn, N. Y.*

Impact Tool: Runs and removes up to ⅝-in. bolts, taps and threads. Can also be used for ordinary drilling, reaming, screwdriving or drawing, hole sawing, wire brushing, masonry drilling and wood boring. Motor runs continuously and cannot be burned out, for as soon as the torque load on the drive spindle builds up to a certain point, rotary action is changed into 2000 sharp blows per minute. Normal clockwise rotation can be reversed by turning the rear end cap. Weight, 6¾ lb. *Servad Inc., Indiana, Pa.*

Hydraulic Crimping Tool: Crimps heavy-duty solderless terminals to stranded or solid wire. Has interchangeable die inserts for eight wire sizes from No. 8 to 4/0. Tool head, in which crimping dies are

inserted, has spring-loaded latch, rotates 360-degrees axially. Standard equipment includes a cradle for use on a bench. Tool can develop 12 tons crimping pressure. *Aircraft-Marine Products Inc., Harrisburg, Pa.*

Drill: "Mighty Midget" has capacity of $\frac{1}{4}$-in. Available in six standard chuck speeds ranging from 600 to 5000 rpm. Features include balanced design, pistol grip handle, polished aluminum housing and a three-jaw geared chuck with rubber-flex chuck key. Operates on either ac or dc, 60 cycles or less. *Stanley Electric Tools, New Britain, Conn.*

Power Plant Equipment

Engine-Generator: Models 3030-X Storm Master, designed to provide standby emergency power. Mounting base consists of four slotted channels arranged in sets of two at right angles to each other, thus providing a transverse mounting system with infinite adjustment within the limits of the frame dimensions. Many different makes and models of engines can be accommodated. *Wincharger Corp., Sioux City, Iowa.*

DC Power Source: Provides constant voltage dc power for all types of automatic welding. Features automatic maintenance of desired preset arc voltage, regardless of fluctuations in arc length; high efficiency and high power factor which subsequently reduces input power requirements; simple installation, operation and maintenance. Models rated at 500 and 1000-amp continuous duty are available for submerged arc, inert gas and manually operated automatic welding. Operate on either 240 or 480-v, three-phase, 60-cycle ac. *Glenn Co., Oakland, Calif.*

Portable Power Supply: For indoor or outdoor use. Completely controlled by electric brakes and clutches. Has Warner pushbutton control system. Operates pumps, compressors, generators, hand tools, and any other machinery which can be transported. Has 1500-w output; operates on either 110-volt current or an independent

guide tissue paper or steel strip . . . *with a breath of air*

Remember the hardest job of web guiding you ever saw? Was it winding up a coil of steel that weighed 100,000 pounds? Was it handling a scrim or tissue as delicate as a cobweb? Was the material loaded with lint or dust or clay or tar? Or moving a mile a minute with runouts of an inch in every lap?

Askania Edge Position Control takes all these problems in stride. And it does a better control job with the everyday problems, too, for the same reasons that make it the only good control for the tough ones.

Askania Edge Position Control is simple. Easy to design into existing equipment. Compact and easy to install.

It's versatile. Controls any material handled in webs: paper, plastic, foils, textiles, metals, rubber, or anything else. Control range can be as great as your own equipment allows.

Askania control is non-contact. All that touches the moving web is a breath of air.

It's powerful. The Askania Jet Pipe puts out enough controlled hydraulic power to handle the heaviest mill rolls with ease.

You can depend on it. Askania Edge Position Control uses the same kind of equipment that heavy industry relies on for the most exacting control jobs.

It's easy to maintain. Almost never needs attention; no special skills necessary to understand it and take care of it.

Let Askania specialists help you analyze your web guiding problem. Write for Bulletin 141. It contains full information about Askania Edge Position Control.

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6-volt system. Normally is towed by a small truck, can be towed by hand for short distances. *Surgie Equipment Corp., New Orleans, La.*

Processing

Metal Treating Furnace: DQ 300 combination draw-quench unit for both scale-free tempering and controlled oxidation tempering from 400 to 1400 F. Fully automatic; has its heating chamber separated from combination cooling chamber and quench tank by a sealed inside door. Controlled endothermic atmosphere provided in both heating and cooling chambers. Parts do not contact air during tempering. After completion of heating cycle, intermediate door opens automatically and chain-driven rods move into heat zone, contact the load with cam-actuated arms and transfer the trays to the quench rack. Tray rack holds work for atmosphere cooling or lowers it for oil quench. For controlled oxidation or blueing, a predetermined quantity of water is admitted into furnace during part of the heating cycle. Size of hearth, 2 ft wide, 3 ft deep, 1 ft high; overall size, 4 ft, 8½ in. wide, 18 ft, 6 in. long, 9 ft, 4 in. high. *Ipsen Industries Inc., Rockford, Ill.*

Core Blower: Model 103 will blow and draw eight cores in a box with a 5½-second machine cycle. Has fully automatic cycle control accomplished with a hydraulically-controlled air timer. Stationary magazine assures perfect alignment of core box. Blower has large throat opening to permit smooth flow of sand from supply hopper, eliminating danger of clogging. *Wm. Demmler & Brothers, Kewanee, Ill.*

Flush-Out Machine: For testing and washing sand and chips from semi-machine cylinder heads. Cylinder head is placed in one side of machine, tripping limit switches. Operator presses buttons, fixture closes, and water pressure is automatically applied to fixture and forced into all unsealed openings in cylinder head. Water is under pressure said to equal a three-inch stream forced 90 ft into the air. *Turner Bros. Inc., Ferndale, Mich.*